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STEAM PUMPS

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STEAM PUMPS

INSTRUCTION PAPER

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STEAM PUMPS.

Principles of Action. A pump is primarily a machine designed for lifting liquids, or for conveying them to a distance through pipes, or both. To accomplish these results may be used the principles of suction, lifting, or forcing or any combination of them; in practice generally the first is combined with one of the other two.

Suction, so-called, is really the pressure due to the weight of the atmosphere acting to force the liquid into a space wherein a partial vacuum has been created by removing a part of the matter that filled it. The amount of pressure available in any instance for forcing in the liquid, depends upon the completeness with which the matter has been removed from the space in which suction is acting, and on the weight of the atmosphere at that place. If all matter could be removed from the suction chamber, a perfect vacuum would be created and the full pressure of the atmosphere would be effective; a condition closely approaching this is created in the barrel of a pump when no air is present and the plunger or part of the liquid is withdrawn.

If there be any air present, it will expand and fill the vacant space, its pressure falling as the volume increases; the pressure available to force in the liquid by suction will be the difference between that due to the weight of the atmosphere and the final pressure in the suction chamber. The greater the amount of matter removed from the suction chamber, the less will be the resulting pressure therein and the greater the difference available for moving the liquid.

The other factor, the weight of the atmosphere, varies with its condition and the altitude as shown by the barometer. The following table compiled from "Kent's Mechanical Engineers' Pocket Book" and "Nystrom's Mechanics" gives barometer readings, the pressure per square inch, and heights above sea level corresponding to each other; the altitude being at 40° latitude and temperature 60° Fahrenheit.

TABLE I.

Barometric Reading in Inches of Mercury.	Atmospheric Pressure in Pounds per Square Inch.	Altitude above Sea Level in Feet.
30	14.72	0
29.75	14.60	232
29.50	14.47	466
29.25	14.35	703
29.00	14.23	941
28.75	14.11	1181
28.50	13.98	1424
28.25	13.86	1668
28.00	13.74	1915
27.50	13.50	2415
27.00	13.26	2924
26.50	13.02	3443
26.00	12.77	3972
25.50	12.53	4511
25.00	12.27	5061
24.50	12.03	5621
24.00	11.78	6194
23.50	11.54	6778
23.00	11.30	7375
22.50	11.05	7985
22.00	10.80	8609
21.50	10.56	9247
21.00	10.31	9900
20.00	9.81	11254
19.00	9.32	12678
18.00	8.82	14179
17.00	8.33	15766
16.00	7.84	17448
15.00	7.35	19240
14.00	6.86	21155
13.00	6.37	23212
12.00	5.88	25433
11.00	5.39	27848

As the weight of a column of water one foot high and one inch square is 0.433 pound, it follows that the atmospheric pressure, which at sea level is ordinarily 14.72 pounds per square inch, can support a column of water as many feet high as 0.433 is contained in 14.72 or 34.2 feet. The height of column which can be supported decreases as the altitude increases. At the top of a mountain two miles high it would be only $9.81 \div 0.433 = 22.7$ feet.

This is the height which could be barely supported if the vacuum were perfect. In practice the height to which water can be lifted is much less, because it is impossible to obtain a perfect vacuum on account of leakage of valves and joints, and the atmospheric pressure must overcome friction in pipes and passages, so that the possible lift is reduced to not over 25 feet at sea level and this only for slow working. It is better to attempt only 20 feet or even less if the pump is to work rapidly. If, however, a high suction lift is imperative, it may be obtained by admitting air with the water in suitable proportions to form a heavy spray so that the weight per cubic foot of the mixture is considerably reduced. In this way a lift of as much as 100 feet may be possible but of course, at the expense of the weight of water pumped per stroke, so that either the pump must be run at a higher speed or a larger pump must be used. The air should be admitted to the suction pipe through several small openings a little above the level of the water in the supply reservoir. The suction must be primed each time the pump is started.

Probably the earliest example of a practical steam pump was that built at Raglan Castle by the Marquis of Worcester in 1630. This employed both suction and pressure, two vessels making the flow continuous. Steam was admitted from a boiler through valve B′, Fig. 1, to the chamber A′, and the pressure forced the water upwards through the valve F′ and pipe E; the valve C′ preventing its return to the reservoir. Meantime the chamber A was shut off from the boiler, and the cold water, condensing the steam, caused a partial vacuum so that water was forced up through valve C and pipe D into the chamber A. When A′ was emptied and A filled, the steam valves were reversed and A′ became filled while A was being emptied.

Practically all pumps, except those to which the liquid comes

under pressure, use the suction principle for filling their cylinders. The suction lift is, however, usually but a few feet except where deep mines are drained, and in the case of air pumps for condensers in which the suction is always kept at the highest possible value. In order to improve the suction action, valves and supply

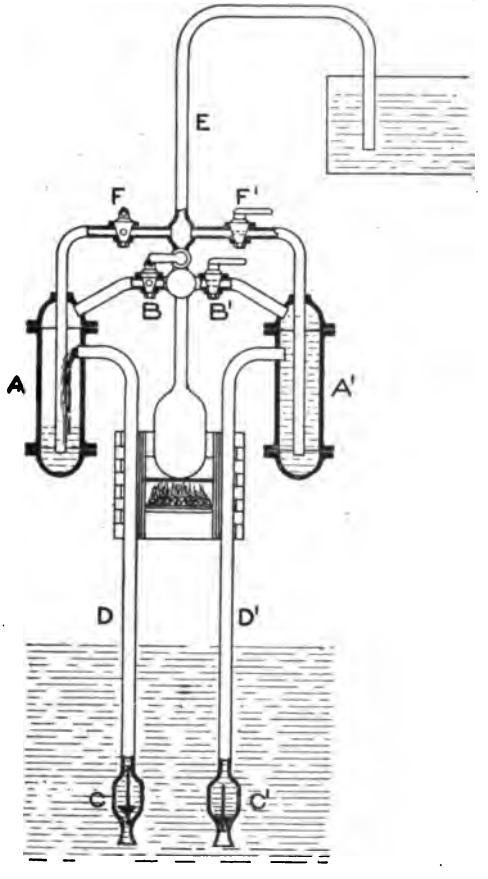

Fig. 1.

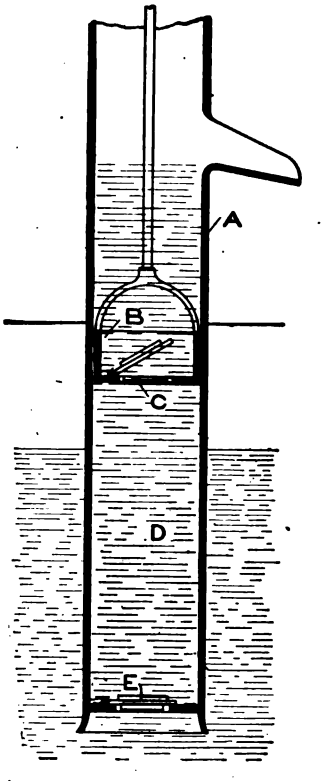

Fig. 2.

pipes must be kept with all seatings and joints absolutely tight.

Lifting. The lifting pump is simply a bucket B, Fig. 2, working in a vertical barrel A and so arranged that the water is caught above the bucket and raised on the upward stroke to the top of the barrel or to a point where an overflow is provided. Usually the bucket is filled on the down stroke by means of a valve

C in the bottom, and if the lift is for any considerable distance, it is necessary to put a foot valve E at the bottom of the inlet pipe to prevent the liquid from sinking in the barrel when the bucket descends. By this means suction also is introduced, since the space D above the foot valve is filled from the reservoir by suction during the up stroke of the bucket. The pump may then be placed at any distance above the reservoir within the limit already stated, and at any distance horizontally; but it should be remembered that the speed of working and height of lift must be decreased if the horizontal distance is materially increased, and that it is always better to make the suction pipe as short as possible.

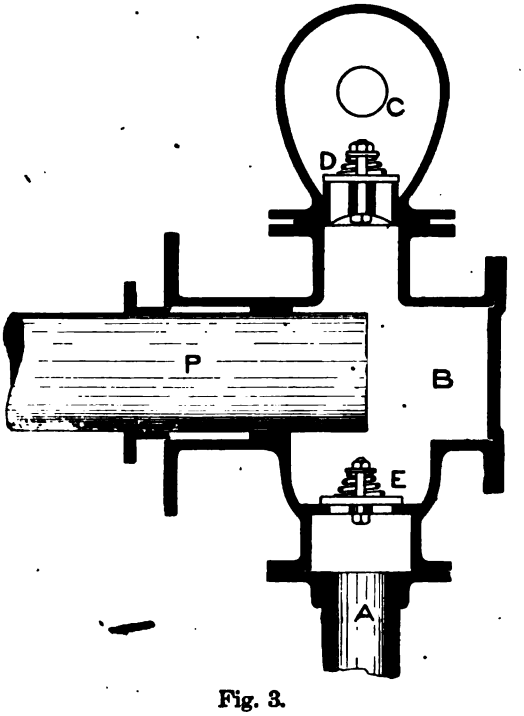

Fig. 3.

Forcing. The suction effect is used in all pumps, at least to fill the barrel, but the lifting principle is uncommon except where the discharge is to be directed from a spout at the top or in the side of the barrel, as in mine drainage and house pumps. If it is necessary to raise the liquid to a point above the pump level, either a piston or a plunger working in a closed chamber is used to force the liquid through discharge valves, against whatever pressure may be necessary, into the delivery pipe; return currents being prevented by the closing of the discharge valves.

As seen in Fig. 3, the plunger P on the outward stroke draws water from the reservoir through pipe A and inlet valves E into the barrel B. On the inward stroke, the valve E will close and,

since liquids are practically incompressible, the discharge valves D will open allowing the water to pass into the discharge chamber and out the pipe C. Fig. 3 represents a single-acting plunger pump, but the action is the same for a piston or for a double-acting pump except that there are two sets of valves. Although the force is exerted on the mass of liquid in a direction lengthwise of the barrel, yet, since pressure is transmitted equally in all directions, the valves will open promptly if located in any position with respect to the barrel.

The pressure against which the pump must act in forcing depends on the weight of the valves, the spring pressure, if any, used to close them quickly and the height to which the liquid is to be raised, or as it is technically termed, the "head," against which the pump is to work. For water, the pressure per square inch increases one pound for every 2.32 feet, or as previously stated one foot of head produces 0.433 pound pressure per square inch. There is no theoretical limit to the height to which a liquid may be raised by forcing, but the speed of starting and stopping the column of water, that is, the number of strokes per minute, must be decreased as the delivery pipe becomes longer, if water-hammer is to be avoided; this is true whether the length be horizontal or vertical. This hammer effect is due to the force necessary to overcome the inertia of the column of liquid and start or stop its motion. If not properly taken into account, either by using slow speeds or providing adequate air chambers to take up the shock, it will always produce destructive results. Wear on the outlet valves also limits the height to which water may be forced, as this wear increases with the head and makes it difficult to keep the valves tight.

Usually no special precautions are required up to pressures of 500 pounds per square inch; above this, care must be exercised up to the limit of 2,000 pounds, which is as high as will be required for lifts or presses except in special cases. Anything above this requires the greatest precautions, not only as to strength of parts, form of valves and packing of joints, but even with respect to quality of castings. Water will sometimes leak through what appears to be solid metal.

The pressure produced by water-hammer is taken advantage of in one special form of pump, **the hydraulic ram**, in which the

TABLE II.

Head in feet.	Pressure pounds per square inch.	Head in feet.	Pressure pounds per square inch.	Head in feet.	Pressure pounds per square inch.
1	.43	46	19.92	91	39.42
2	.86	47	20.35	92	39.85
3	1.30	48	20.79	93	40.28
4	1.73	49	21.22	94	40.72
5	2.16	50	21.65	95	41.15
6	2.59	51	22.09	96	41.58
7	3.03	52	22.52	97	42.01
8	3.46	53	22.95	98	42.45
9	3.89	54	23.39	99	42.88
10	4.33	55	23.82	100	43.31
11	4.76	56	24.26	101	43.75
12	5.20	57	24.69	102	44.18
13	5.63	58	25.12	103	44.61
14	6.06	59	25.55	104	45.05
15	6.49	60	25.99	105	45.48
16	6.92	61	26.42	106	45.91
17	7.36	62	26.85	107	46.34
18	7.79	63	27.29	108	46.78
19	8.22	64	27.72	109	47.21
20	8.66	65	28.15	110	47.64
21	9.09	66	28.58	111	48.08
22	9.53	67	29.02	112	48.51
23	9.96	68	29.45	113	48.94
24	10.39	69	29.88	114	49.38
25	10.82	70	30.32	115	49.81
26	11.26	71	30.75	116	50.24
27	11.69	72	31.18	117	50.68
28	12.12	73	31.62	118	51.11
29	12.55	74	32.05	119	51.54
30	12.99	75	32.48	120	51.98
31	13.42	76	32.92	121	52.41
32	13.86	77	33.35	122	52.84
33	14.29	78	33.78	123	53.28
34	14.72	79	34.21	124	53.71
35	15.16	80	34.65	125	54.15
36	15.59	81	35.08	126	54.58
37	16.02	82	35.52	127	55.01
38	16.45	83	35.95	128	55.44
39	16.89	84	36.39	129	55.88
40	17.32	85	36.82	130	56.31
41	17.75	86	37.25	131	56.74
42	18.19	87	37.68	132	57.18
43	18.62	88	38.12	133	57.61
44	19.05	89	38.55	134	58.04
45	19.49	90	38.98	135	58.48

force produced by the inertia of a large column of water suddenly checked is utilized to raise a small column against a high pressure. The working of the apparatus is seen from Fig. 4. Water flowing from pipe F into the chamber A will escape through valve V (closed in the figure) to discharge pipe D; but, as the velocity of flow increases, valve V will be lifted and closed checking the flow of water. The force thus developed will open valve V′ against the pressure of the spring and the head of water in the small pipe E, and some water will be forced into chamber B and up pipe E or into the air chamber C compressing the air. Water will pass into B until the force due to water hammer is so decreased that valve V′ closes and valve V opens once more, when water will again overflow and the process be repeated. The head against which the ram will work may be almost any amount, but the ratio of water raised to that wasted at the overflow will decrease as the head on valve V′ increases. In any case, the ratio will be small and the ram wasteful, so that its use is advisable only where the supply of water is almost unlimited, other power is difficult to procure, or the amount to be raised is very small. Table III gives the relation of efficiencies and ratio of lifting to forcing heads, which according to D. K. Clark, may fairly be expected with a ram of good design.

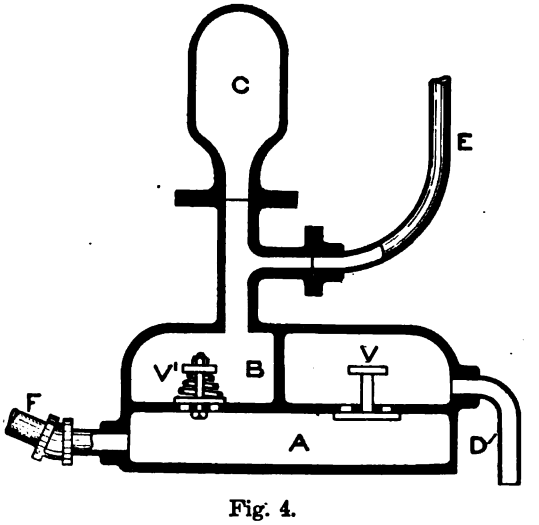

Fig. 4.

TABLE III.

Ratio of lift to fall	4	6	8	10	12	14	16	18	20	22
Efficiency per cent	72	61	52	44	37	31	25	19	14	9

The ram should be arranged with a drive pipe whose length is

at least five times the fall, in order to give a long column of moving water and increase the water-hammer. The diameter of drive pipe is usually twice that of the lift pipe.

KINDS OF PUMPS.

In classifying pumps, the division may be made according to any one of several methods: As to the principles of action, the details of construction, the means of driving, or the purposes for which they are used. With respect to principle of action the classes may be:

1. Lifting,
2. Forcing,
3. Combinations.

With regard to the details of construction:

1. Jet,
2. Rotary,
3. Centrifugal,
4. Reciprocating.

With regard to uses, the division is difficult, since the same pump is frequently used in different places for different purposes. Such classes as tank pumps, boiler-feed pumps, air pumps and mine pumps are fairly definite, but the details vary so much with different makers that it is impossible to draw sharp distinctions.

From the discussion of the principles of action we know that practically all power pumps are of the third class, that is, work by a combined lifting action due to suction and forcing. But the second classification is yet to be considered.

The Jet Pump. The first variety, the jet pump, is perhaps best known in the form of the injector, but the principle is worthy of a much wider application than it now has. In the form of a steam or water ejector it may be used for lifting water by suction and raising it to a considerable height, while its simplicity and ease of application give it a large field of usefulness. A convenient form for an ejector is shown in Fig. 5. The propelling jet may be either steam or water under pressure entering through the valve A and flowing from nozzle D which should be made with a decreasing taper like a hose nozzle. If steam is used, it will be necessary to provide a small injection pipe with valve for supplying water to condense the jet and create a vacuum at the start, otherwise the steam will blow through the apparatus and will not

lift the water in the suction pipe. In this case, the heat energy of the steam will be converted into velocity, and will be available for drawing and forcing the water so that it can be raised to a considerable height. The efficiency of a well-designed steam ejector should be nearly equal to that of an injector and will, like that, depend on the temperature of the water and the height of the suction and lift.

In pumping into a boiler, the efficiency of an injector is nearly 100 per cent because nearly all the heat energy which is lost in friction goes to heat up the feed water and is therefore utilized;

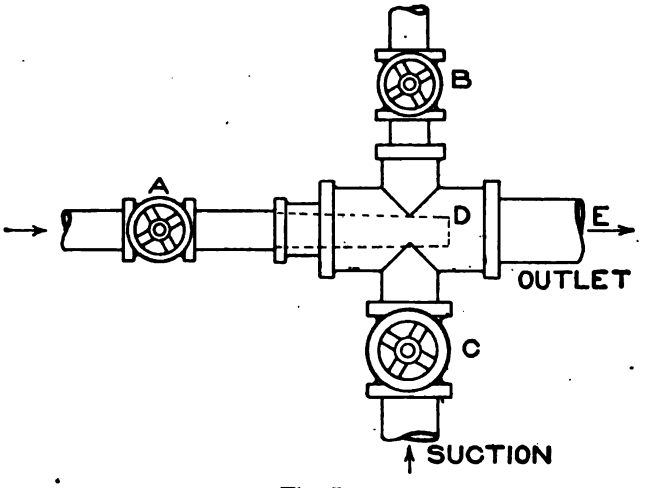

Fig. 5.

but for other purposes, the injector or ejector is wasteful of heat and cannot be considered economical. Its convenience, however, is a strong point in its favor, and where the amount of water to be lifted is not large, or if it is to be used only occasionally, it will often pay to use it rather than to install a more elaborate and costly pump.

When water under pressure is used as driving power, it is not necessary to have the inlet at B. The flow of the water from the nozzle will carry along the air and create a vacuum at D which will draw the water up the pipe C. There is no heat energy available as in the case of steam, so that the water can furnish only such an amount of power as is due to its weight and the velocity with

which it leaves the ejector nozzle; and, as the efficiency is low, not over one fourth of the energy furnished to the ejector at A will be available to move water at E. For a water injector it is well to proportion the pipes so that the head on A times the area of D equals twice the suction and lifting heads combined multiplied by the area of C; and to make the area of E equal to the area of D plus the area of C, if the apparatus is to be used as a water ejector. For steam, a $\frac{1}{2}$-inch pipe at A for a 2-inch at C and a $2\frac{1}{2}$-inch at E has been found to work well when the head on E was small, and if the water is under city pressure a $\frac{3}{8}$-inch pipe at B.

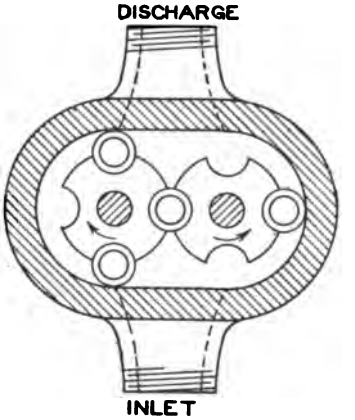

Fig. 6.

It is well to remember that the energy of a jet of either steam or water depends more on the velocity than on the weight flowing, since the energy available per second equals $\frac{W v^2}{2g}$ where W is the weight flowing per second, v is the velocity in feet per second and g is the acceleration due to gravity; therefore, for good efficiency, it is best to use high pressure whenever possible.

Rotary Pumps. These have been but little used because, as usually designed, their efficiency has been low. They are, however, useful in many cases and are well adapted for moving large volumes of liquid at low pressures. This pump moves the liquid by catching it between the lobes of the revolving propellers and the casing of the pump and sweeping it around from the suction pipe to the discharge; its return being prevented by the obstacle of the lobes and cylinders in the center. All are made on the same principle, the difference being in the design of the lobes and cylinders.

These are of many forms, but two will be sufficient to illustrate the principles. Fig. 6 is the Berrenberg; here the lobes are made by two pieces of tube fastened opposite each other. The tubes extend from the semi-cylindrical openings and make a joint with the other lobe by fitting in as shown. The axles are geared

together outside the casing and are supported by bearings inside and outside, which are tapered to allow taking up wear and keeping the shafts true. The curves in Fig. 7 show the results from a series of tests made on a 2-inch pump by Mr. R. A. Hale and recorded in Barr's "Pumping Machinery." These show that the efficiency of the pump increases with the pressure against which it is working up to a certain point, and then falls off, while comparison of the curves at different speeds shows that

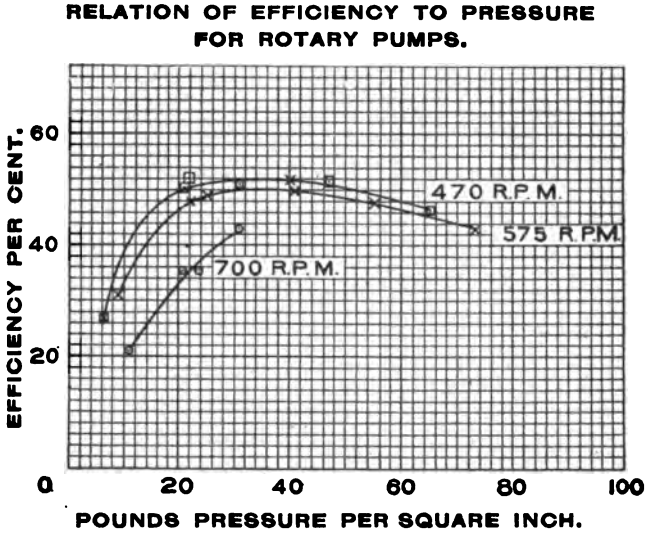

Fig. 7.

the efficiency becomes less as the speed increases. This is apparent because the loss from friction is not doubled by doubling the head, while the amount of water will be the same per revolution, no matter what the pressure may be. The exception to this is due to slip which increases as the pressure becomes greater. Hence the useful work will increase up to that point where the loss due to leakage and friction more than balances the increase of work due to the increased head. The same reasoning applies to the speed, except that in this case the friction work will increase in proportion to the speed, so that the gain from the added work due to the increase of speed is not as great as from increase of head.

Another style of rotary pump is shown in Fig. 8 in which the two pistons are so designed that their bounding surfaces are always

in contact, thus forming the joint. In this type it is important that the surfaces be of the correct shape, otherwise a pocket will be formed in which water will be caught. This water will be compressed as the volume of the pocket becomes smaller. As water is practically incompressible, it must either leak past the joint or burst the pump. In order to prevent this, a port is sometimes cut in the casing to allow the water to bypass and relieve the pressure. If, however, the outline of the lobes is made of the right shape, there will be no compression and the ports will not be nec-

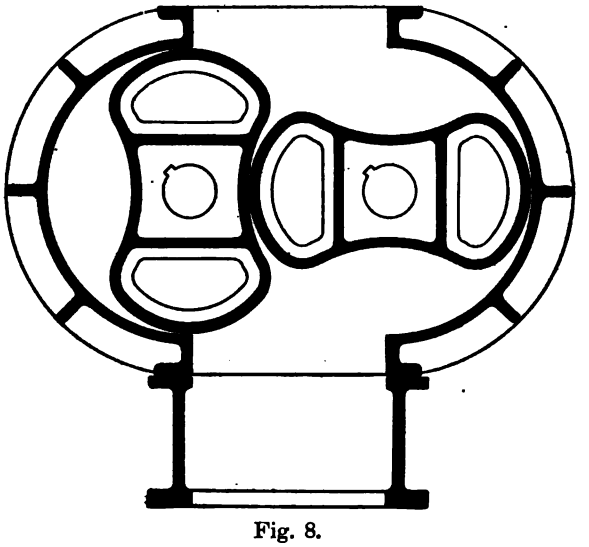

Fig. 8.

essary. It has been demonstrated that the curves to use for the outline of the lobes are the cycloids, as they can form the whole outline and will make a smooth surface which will not create pockets at any part of the revolution. The involute curve has sometimes been used, but it is not good as it does not return on itself, and must therefore be cut off by the arc of a circle on the end of the lobe, leaving corners which form compression pockets.

With respect to the effect of the speed on the volume of water pumped, the experiments above noted showed that, within the stated limits, the volume varies nearly as the speed, and that at the same speed, the volume is a little less at high pressure than at low, but does not fall off rapidly as the pressure increases. The data, however, is not complete enough to be conclusive.

The rotary pump is often used for fire purposes, and where portability is of more importance than efficiency, or if first cost is a great consideration. It can be run at high speed, and has a large capacity for the space occupied, but it is not suitable for large or permanent work except at very low heads.

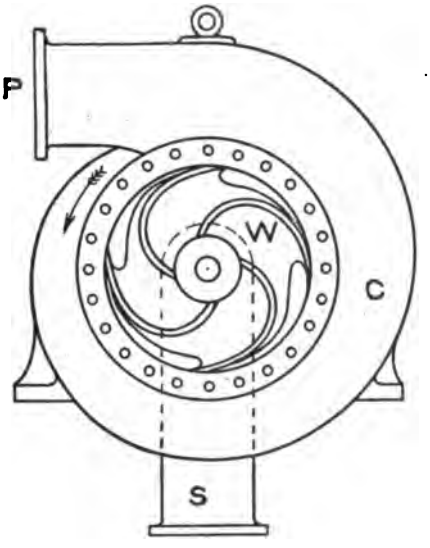

Fig. 9.

Centrifugal Pumps. These operate by the familiar force which throws the mud from the carriage wheel and the water from the lawn sprinkler. A wheel W (Fig. 9) with curved blades revolves in a casing C and, when the casing is filled with water, throws it to the outside thus creating a suction at the center and drawing up water through the suction pipe S. The velocity and whirling motion given to the water send it to the outside of the casing and out through a delivery pipe P. As seen in Fig. 9, the casing increases in size up to the outlet P, as the volume of the water to be carried will increase in that way. The blades are given a backward inclination at the wheel center so that water may enter without shock, and the angle which the blades make with the radius is increased towards the outer end so that the water gradually attains some rotary velocity and great radial velocity due to the wedging action as it passes outwards; on leaving the blades it has high speed. If the blades are radial at the outer end, the discharge will be greater for a given speed, but it will not be possible to reduce the speed much without stopping the action altogether; also the efficiency will be less.

Rankine states that the curve of the blade should be an involute of a circle, that is, a curve drawn by a point in a string as it is unrolled from the circle as shown in Fig. 10. The method of drawing the involute is as follows: Let BC be the inner bounding circle

of the blades and B the point at which a vane is to start. Draw a radius of the circle through B and through the same point at an angle of 40 degrees with the tangent, another straight line BE. Draw a concentric circle tangent to BF, which is perpendicular to BE. If B were a point in a string, and this were unrolled from the circle, the point B would trace an involute BGH. For the first form of curve mentioned, this would be continued to the outer bounding circle where it should make an angle of about 15 degrees with the tangent, as LHM. For the Rankine blade, the reverse curve HKA should be drawn, usually the arc of a circle radial at the point H. In order to work without shock, the angle of the blade with the tangent at the inside end B should be such that its tangent equals the radial velocity of the water divided by the circumferential velocity of that point on the wheel; it is often made 40 degrees, which is found to be satisfactory for the average case. The angle at the outside should be 15 degrees if there is no diffusing chamber, or if the discharge pipe is not of increasing diameter, in order to gradually lessen the velocity of the water after it leaves the wheel. But this will necessitate a high rotary velocity of the blades, and it is sometimes better to use a diffusing chamber (or an increasing discharge pipe), and make the blades of the Rankine form, running the wheel at a lower speed unless the conditions make it necessary to be able to run at a given medium speed for a partial discharge. The casing should be shaped to give the water a gradually increasing velocity from the time that it enters the suction pipe until it reaches the circumference of the fan, and then a decreasing velocity until it leaves the discharge pipe. The spiral casing is often recommended because the centrifugal force given by the rotary motion is radial, not tangential, and the water must be

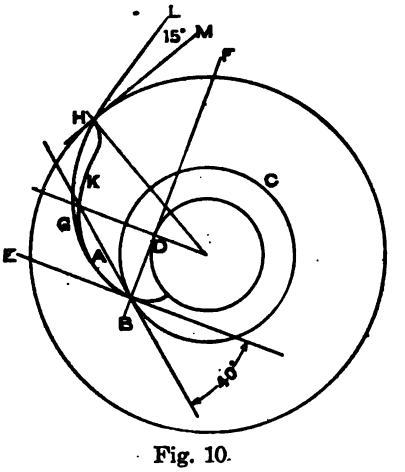

Fig. 10.

given time to change its direction or it will spend much of its energy in eddies and useless work.

Three styles of blades are shown in Figs. 11, 12 and 13. Fig. 11 is the form used for small sizes and for thick liquids; Fig. 12

Fig. 11.

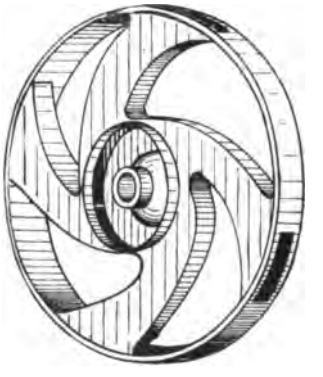

Fig. 12.

is a hollow-arm type used in large pumps; it has the advantage that the water is thrown outward without any churning motion, and that there are no dead spaces. Fig. 13 is the style used for dredges and has the advantage that the sand is kept from grinding between the blades and the casing yet large openings are free for the passage of sand and mud.

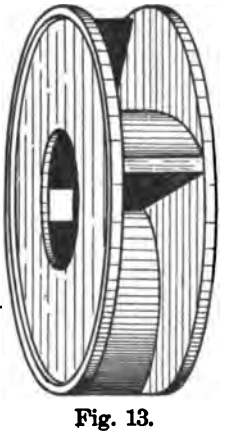

Fig. 13.

It is possible to raise water to any height up to 100 feet by increasing the speed of the pump, but it has been shown by experiment that this form is not suitable for lifts of over 25 feet, although up to 35 feet the efficiency is fairly good. As it is at these low lifts that the reciprocating pump is least efficient, this is clearly the field of the centrifugal pump.

The speed is increased as the height of the lift increases, but not in exact proportion, because the increase of speed for an increase of 10 feet in lift is greater at low than at high heads. Fig. 14 shows the relation between head and speed in two pumps of the same make, the No. 2 having a capacity of 100 gallons per minute at the speeds given

and the No. 6 a capacity of 1,200 gallons. The point at which the curves cut the axis of speed shows that the No. 2 pump will fail to work at a speed below 250 r. p. m. and the No. 6 below 125 r. p. m. The pump may be set in either a vertical or a horizontal position and may have the suction on one side or both. However, it is better on both so that the forces due to lifting the water in the suction pipe are balanced and no end thrust produced. It is necessary to have some means of filling the pump casing with

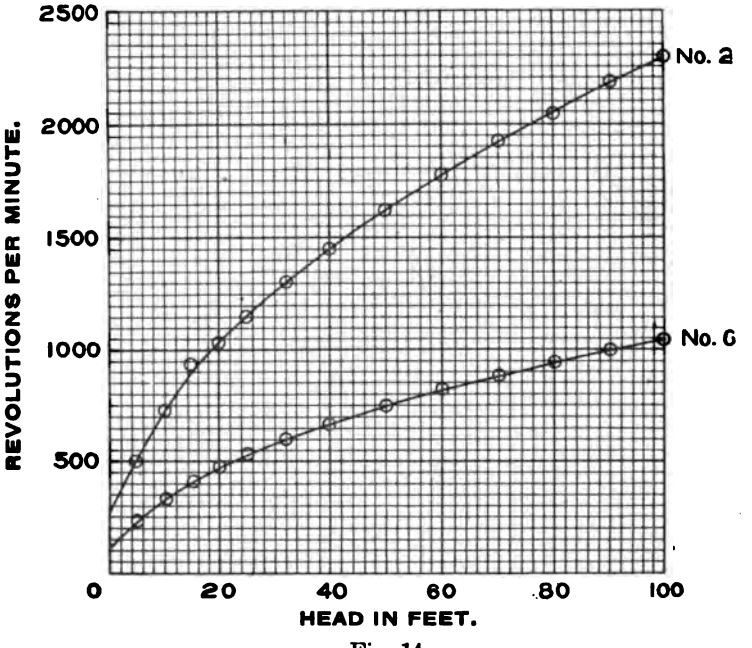

Fig. 14.

water, that is, "priming" it, as the suction will not start until the pump is so primed. If water under pressure is not available, it is necessary to install a steam injector to fill the casing, or provide a foot valve which will be tight enough to hold the water in the suction pipe at all times.

Makers recommend that a flap valve be installed at the top end of the discharge pipe also, to assist this action. Fig. 15 shows an arrangement which is satisfactory and also gives an idea of the shape and relative proportions of a standard centrifugal pump.

Lifting Pumps. The only pumps which work entirely by

lifting are the old-fashioned chain devices in which a series of discs mounted on a chain are drawn, by a sprocket wheel, up through a tube bringing the water with them. In this form the motion is always in one direction, and there is a purely lifting action. But it is common to speak of the pump which delivers its stream by lifting, as a lifting pump even though it is filled by the action of suction. These are used largely in mines and for domestic purposes. The principle has been shown in Fig. 2. The disadvantage of this style is that the pump bucket must be not over 20 feet above the water and the rest of the lift must be made by the rise of the bucket, necessitating either a long rod, or pumping in relay.

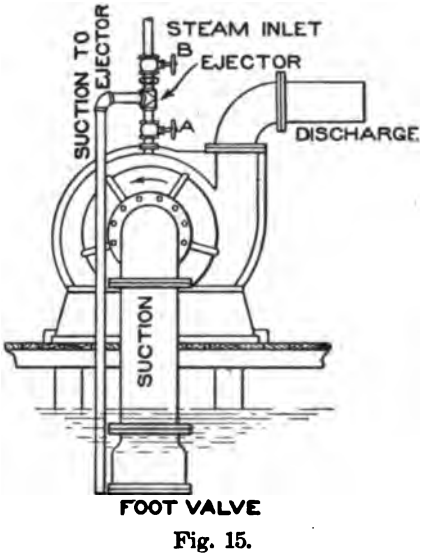

Fig. 15.

The greatest use of this class was in the Cornish mines in the time of Watt when the engines were placed at ground level, and the rods made of a length sufficient to reach to within 20 feet of the bottom of the shaft. In some cases these pump rods were as much as 480 feet long, and the design of the great pieces was a difficult problem. The walking-beam engine was used largely to drive the pumps, and the whole machine was a huge and wasteful affair.

At the present time, the long pump rod is used to some extent, but it is driven by a vertical cylinder set over the top of the mine shaft.

The size of the pump cylinder depends upon the volume of water to be raised and the speed of the pump. When heavy masses are to be moved, as is the case when long rods are used, the speed must, of course, be slow. For the great engines of Watt's time it was about 15 strokes per minute, and in modern pumps it is 25 to 35 strokes per minute. The cylinder must be made of such size that at this speed it will lift the required amount of water.

For example, if it is desired to raise 100 gallons per minute, and the pump is to run at 30 strokes, the volume to be raised is $100 \times 231 = 23{,}100$ cubic inches, and the volume per stroke is $23{,}100 \div 30 = 770$ cubic inches. A usual value for the stroke of pumps of this size is 16 inches which (if there were no allowance for leakage) would make the area of the cylinder $770 \div 16 = 48.13$ square inches. The allowance for leakage or slip should be, for a pump of this size and speed, about 3 per cent. This would, of course, increase with high speed and decrease with large size. Allowing 3 per cent slip, the area should be $48.13 \times 1.03 = 49.58$ square inches which is that for a circle of about $7\frac{5}{16}$ inches diameter, so that 8 inches would be used and the area would then be 50.26 square inches.

To find the force needed to lift this bucket, it is necessary to know the height to which the water is to be raised. Suppose this to be 200 feet; then the pressure per square inch will be $200 \times .433 = 86.60$ pounds and the force needed, $86.60 \times 50.26 = 4{,}352+$ pounds To this must be added the weight of the bucket and rod as both must be lifted at each stroke by the driving head at the top of the shaft. The rods are usually made of wood with the sections fastened together by iron couplings. The proper size for the above pump is 4 inches diameter, which for a rod 200 feet long would weigh, at 45 pounds per cubic foot, $45 \times \frac{\pi \times 4^2}{4 \times 144} \times 200 = 785$ pounds (about), and the bucket and fastenings will easily make this 1,000 pounds; the whole weight to be lifted will then be 5,352 pounds, which, at 30 strokes per minute, will call for the expenditure of

$$\frac{5{,}352 \times 30}{33{,}000} \times \frac{16}{12} = 6.5 \text{ H. P. (nearly)}$$

This would be the power necessary to run the pump aside from the friction of the machinery.

Although, at the present time, high lifts are largely made by using a forcing pump at the bottom of the shaft, this was not known at the time of the introduction of the pumping engine in the Cornish mines, and when it was necessary to raise water from a depth greater than could well be handled in one lift, it was accomplished by using pumps in relays. One engine would be

installed at a depth of some 240 feet from the bottom of the mine, with its pump cylinder within 20 feet of the bottom, and would raise the water into a reservoir from which it would be taken by another pump and so on. Instances are recorded in which pumps raised 4,614,800 gallons per 24 hours, having pistons 24 inches in diameter and a stroke of 10 feet.

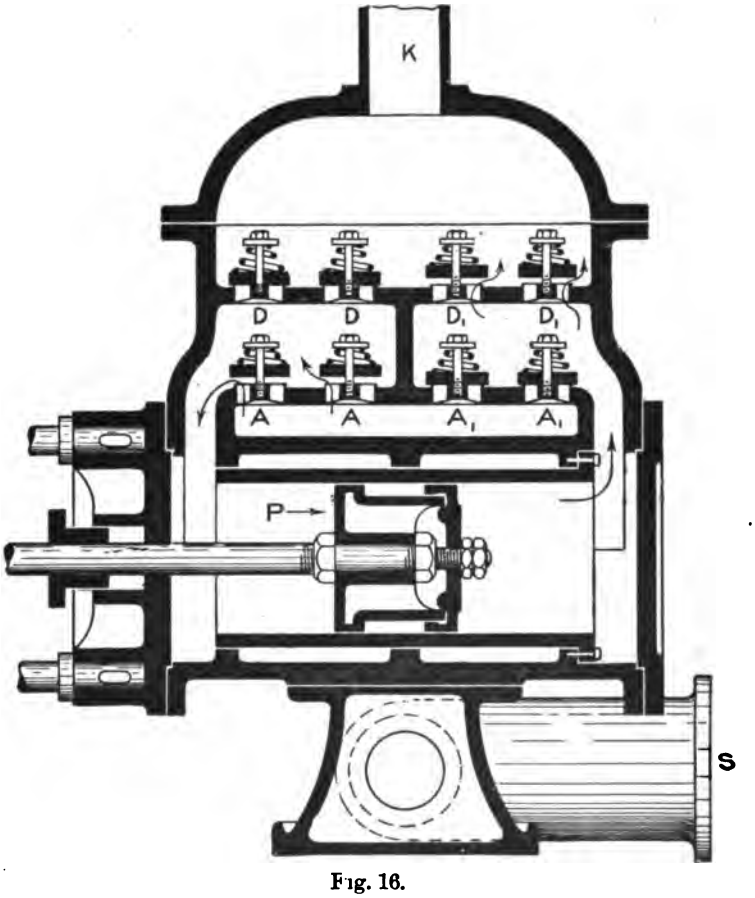

Fig. 16.

Forcing Pumps. For raising water to a height greater than can be conveniently overcome by lifting, or when the discharge is desired at a distance from the pump, it is better to use the forcing type. The simple principle of this type is shown in Fig. 3. There are, however, many modifications in common use. It is usual, if the lift to the pump be great, to install a foot valve at the bottom

of the suction pipe, as well as to use admission valves to the pump chamber so that the suction pipe will be kept filled at all times, and the tendency for the water to drop out of the pump chamber, while the suction valve is closing, be removed.

The most common form of forcing pump is the double-acting piston type as seen in Fig. 16. As the piston P moves to the right, it will create a suction in the left-hand end of the cylinder, drawing in water through the valves AA and the suction pipe S. At the same time pressure will be exerted in the right-hand end of the cylinder and water will be forced out of valves D_1D_1, into the discharge chamber and up the pipe K. On the reverse stroke, water will be drawn in at valves A_1A_1 and forced out of DD. The piston must, of course, be packed water tight.

As in the case of lifting pumps, the volume to be raised and the speed at which the pump can be run determine the size of the piston; but the speed is usually greater than in the lifting type. The pump must not be placed over 20 feet above the level of the water which it is to raise, hence for a high head the greater part of the lift must be obtained by forcing against the head above the pump. It is a common rule that the travel of a pump piston should not exceed 100 feet per minute, and, taking the same problem as for the lifting pump this gives, $\frac{23{,}100}{100 \times 12} = 19.25$ square inches as the area of the piston. At the increased speed, the slip will be greater, hence it will be necessary to allow 5 per cent or an area of $19.25 \times 1.05 = 20.21$ square inches corresponding to a diameter of $5\frac{1}{16}$ inches.

The head would be the same, hence the total pressure per square inch would be the same as for the lifting pump, but the power would be increased. The total force on the piston will be $20.21 \times 86.60 = 1{,}750.1$ pounds. At 100 feet per minute this will require $\frac{1{,}750.1 \times 100}{33{,}000} = 5.30$ horsepower, which is less than before because the weight of the pump rod is not lifted at each stroke. If the rod be counterbalanced by a weight at the opposite end of a walking beam, the lifting pump will have a slight advantage owing to the smaller slip.

The pump shown in Fig. 16 is double acting, that is, it

works on both strokes, but pumps, especially of the vertical type, are often made single acting to suck on the up stroke and force on the down; these have only half the capacity for the same size as the double acting.

The great difficulty with the piston is in keeping the packing tight or of knowing when it is leaking; it is also difficult to

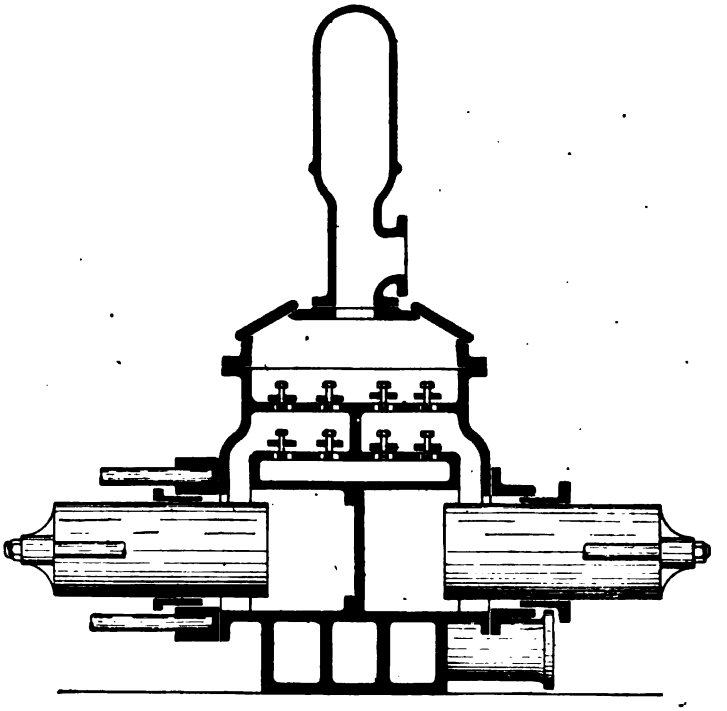

Fig. 17.

repack as the pump must be dismantled in order to get at the piston. For this reason the plunger pump is often preferred and is used either single acting with stuffing box around the plunger as in Fig. 3, double acting with two plungers as in Fig. 17, or double acting with one plunger as in Fig. 18. The advantage of this last over the piston is that for high pressures it is more easily kept tight, and for gritty liquids the wear is taken by the bushing which is more easily and cheaply replaced than a lining to the cylinder. The piston pump is more compact; but the plunger

does not require a bored cylinder so that the first cost is not materially different.

The two-plunger pump with outside yoke is still larger than the single-plunger double-acting type but it can be easily repacked, can be kept tight against high pressures and any leaking is instantly detected. For these reasons, it is a favorite. The double-acting pump, not only has greater capacity, as above mentioned, but, on account of its continuous action produces a much steadier flow than the single-acting type, so that it is generally preferred.

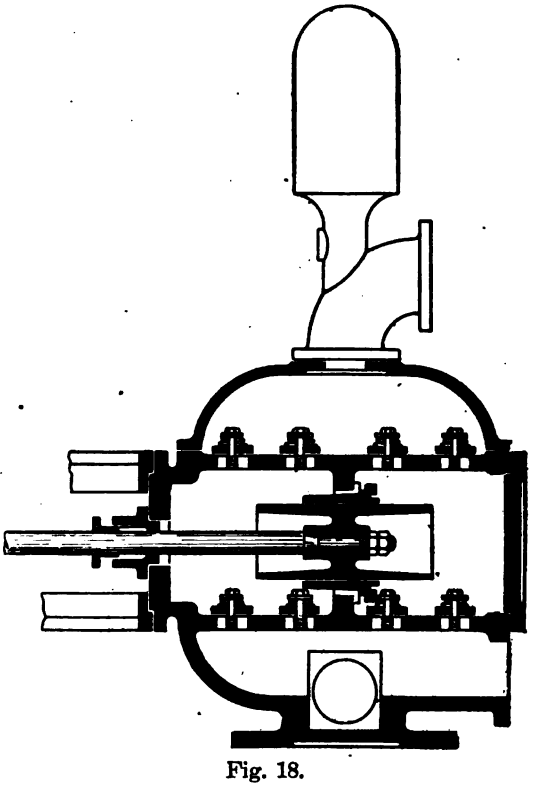

Fig. 18.

A modified form of Worcester's Engine, Fig. 1, is seen in the pulsometer or hydrotrophe, Fig. 19, which is still used to a considerable extent where extreme simplicity is desired. The ball B rolls so as to close first A′ and then A; with the ball in the position shown and the chamber D filled with steam, water will be drawn up through F by the vacuum caused by condensation of the steam. Meantime steam will enter D′ through A and force the water out through the check valve G. This will continue until the steam in D is all condensed, when the rush of the water, due to its inertia, will drive the ball valve B over against port A, thus reversing the action. It is entirely automatic and, as the machine is the acme of simplicity and may readily be made with large openings, it will handle dirty

liquids with a minimum of attention. Its usual heat efficiency is 1.2 to 1.5 per cent, but this is not as objectionable as it seems, since small reciprocating steam pumps often do little or no better. C is a suction air chamber connected with the opening E to equalize the flow of water.

VALVES.

Pump valves must be so made that they will open and close quickly and fully with little friction, so that there shall be no obstruction to the passage of the water, yet little leakage of water past the valves after the reversal of motion of the piston and before the valves have seated. Quick opening requires a light valve with slight motion, but quick closing demands either a heavy valve or a spring to hasten its action. In older pumps it was the custom to use one large valve with a lift sufficient to give the required passage; the valve was then heavy enough to close of its own weight, but was clumsy and hard to fit to a seating. In modern practice the required area is divided among several small valves so that each one is easily and cheaply renewed in case of accident or wear, and quick closing is obtained by the use of springs on the spindles of the valves.

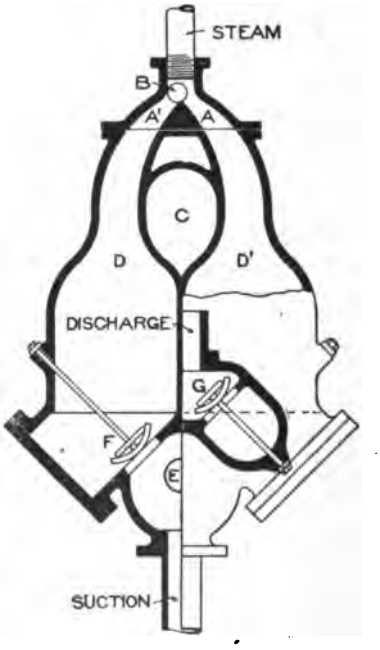

Fig. 19.

The natural classification of valves is by the motion and by the method of securing a tight joint; according to the motion there would be division into flexible, hinged, poppet and ball; according to the seating, into cushioned and ground.

The Flexible valve is the oldest type, and is, in some ways, the simplest. It is generally made of rubber, as this material gives the greatest flexibility with the best seating. The valve is made as a solid disc, the opening being obtained by the flexibility of the

valve itself, and is usually of the style shown in Fig. 20. The seat is made with radial grids to take the pressure of the water, the openings between the bars being made equal to the thickness of the disc; this thickness depends in turn on the diameter of the disc, and is often made as given in the following table:

Diam. of disc. Inches.	Thickness of disc. Inches.	Diam. of hole in center. Inches.
2	$\frac{3}{8}$	$\frac{1}{8}$
$2\frac{1}{2}$	$\frac{7}{16}$	$\frac{1}{8}$
3	$\frac{1}{2}$	$\frac{9}{16}$
$3\frac{1}{2}$	$\frac{5}{8}$	$\frac{11}{16}$
4	$\frac{5}{8}$	$\frac{11}{16}$
$4\frac{1}{2}$	$\frac{3}{4}$	$\frac{13}{16}$
5	$\frac{3}{4}$	$\frac{13}{16}$

The valves of this type most commonly used are the 3-inch and the $4\frac{1}{2}$-inch as the larger sizes are not found durable. The 3-inch is the best for ordinary use. The hole in the center is made $\frac{1}{16}$ to $\frac{1}{8}$ inch larger than the spindle to allow of free motion; to avoid sudden bending of the rubber, the guard is placed to give the disc a chance to rise about $\frac{3}{16}$ inch. When it has risen so far, it bends around the guard and thus gives a free opening for the passage of the water. The guard should have a diameter equal to that of the disc less twice the thickness of the disc, and the radius of curvature should be equal to the diameter of the disc. This type of valve is not well suited for high pressures, 250 feet head being about the limit for the ordinary style. If, however, there are vanes arranged below the disc so that a rotary motion is given to the valve, it is found that there is no difficulty in keeping the valve tight for heads up to 490 feet. For these pressures, it is not uncommon to use discs of a thickness up to an inch, but only for large sizes. The lap of the valve over the edge of the seat should be from $\frac{1}{4}$ to $\frac{1}{2}$ inch according to the thickness of the disc and the pressure to be carried.

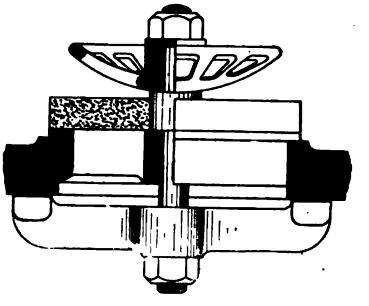

Fig. 20.

Leather valves have been used to some extent in the form of a simple flap fastened at one side and free to rise at the other; this makes the simplest form of hinged valve and, when weighted with a metal disc on either side, has been much used for small hand pumps. It is not suitable for severe service.

The Hinged valve, as its name implies, is made to swing about a pivot at one side, but should be designed to lift from its seat before it begins to swing. The disc is made of metal but often with a facing of leather or rubber for slow speeds; for high speeds these linings will not stand the wear. The lift should be about $\frac{1}{8}$ to $\frac{1}{4}$ inch before the valve begins to turn about the fulcrum. The angle to which the valve lifts is usually made 30 degrees, and should never be over 60 degrees, in order to ensure quick opening and closing. The motion is limited by a stop on the seat or on the disc of the valve as seen in Fig. 21. The hinged valve is slow in

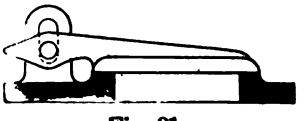

Fig. 21.

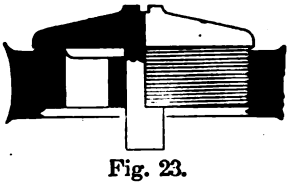

Fig. 23.

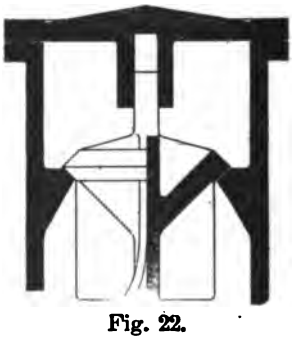

Fig. 22.

action and is not suitable for thick liquors or high pressures. In designing, the area allowed should be as follows:

Piston speed, feet per minute.....	100	125	150	175	200
Area of waterway, per cent of plunger area.	40	50	60	75	100

For higher speeds or for high pressures, it is best to use some form of positively driven valve, and, if such liquors as sewage or syrups are to be handled, the areas found from the above table should be increased about 50 per cent.

The most common form of valve in use at present is the **poppet** which rises straight up from the seat and gives a free opening with a small motion. The seat is made either flat or at an angle of 45 degrees as the designer may choose, the beveled

seat being more difficult to construct but retarding the flow of water less than the flat seat which necessitates a double turn in the current. The width of the beat or edge of the seat against which the valve rests is made $\frac{1}{4}$ inch for large valves and $\frac{1}{16}$ inch for small ones. The flow of the water through the valves should be at a velocity of not over 200 feet per minute, and the lift must be so proportioned that the passage by the valve will be as large as the area of the valve opening. A valve with flat seat need not have over $\frac{1}{4}$ inch lap even for heavy pressures, but the joint must be carefully made to avoid leakage.

A common form of **mitre valve** is shown in Fig. 22 and of **flat seat** in Fig. 23. The vanes on the bottom of the disc which keep the valve centered when it rises are often given an inclination of $\frac{1}{2}$ inch in 6 inches so that the rush of the water will give the valve a rotary motion, removing all dirt and keeping the wear on the beat even. For the flat beat valves this is not always possible as they are closed by a spring and the friction prevents them from revolving.

Fig. 24.

In order to give an area by the valve equal to the area of the seat, the lift must be one quarter the diameter of the seat opening. For heavy pressures this would make too large a lift, as the slip on closing would be excessive; for this reason the lift is made not over $\frac{1}{4}$ inch and the number of valves increased as may be necessary. The seat is usually made separate from the valve deck and screwed into place; when this construction is used, the guide for a valve (if there is one) should be made a part of the seat so that the valve is self contained. The guides may be in the form of wings on the bottom of the valve disc, a spindle running through a hole in the seat or a spindle projecting upward through the valve; the last being most common.

The discs may be of metal or hard rubber. For hot liquids metal must be used, but for cold liquids hard rubber or composition is more common, usually with a metal cap to keep the disc

in shape. The spring used for closing the valve should be cylindrical rather than conical, as the strain on the latter form comes largely at the small end of the spring and the wire is likely to break there. A design to be recommended is that of Fig. 24 which is described by Barr; see "Pumping Machinery," p. 57.

Double beat valves and those with multiple ports are not usually necessary but are sometimes used for large work.

Ball valves are often used for low pressures. For these the diameter of the ball is made $1\frac{1}{3}$ times that of the beat and a cage is provided so that the ball shall always be kept central. The ball is of gun metal usually with a rubber casing or, for cold water, sometimes wholly of rubber. They are common in pumps used for mines and oil wells and often in connection with cup leather packings for the piston. If the water is gritty or has impurities, the valve seat should be raised above the deck so that there will be room for the sand to settle below the seat, as seen in Fig. 25.

Fig. 25.

In all valves which have metal for both discs and seats, it is important that they should both be of the same material, as otherwise there is likely to be galvanic action which will cause rapid deterioration. Usually, however, it is better to use the valve with a separate disc of composition as this avoids all possibility of electrical action and makes the refacing of the valve unnecessary; the replacing of the old disc by a new one being all that is needed when the valve becomes worn.

For valves which are to be returned by a spring, it is best to use five coils in the spring and to have the wire of spring brass of the size given in the following table:

Diameter of valve inches	2	$2\frac{1}{2}$	3	$3\frac{1}{2}$	4	$4\frac{1}{2}$
No. of wire B & S	12	12	10	10	8	8

In this style, it is well to use for the guiding spindle a stud screwed solidly into the valve seat or made a part of the casting, and fasten the valve by a nut at the top of the stud, secured by a pin. This is usually better than to use a stud screwed into the

valve seat. If the latter is used, it is likely to work loose and may cause much damage before the trouble is discovered; it is also more difficult to remove when repairing the valve.

The material for the valve depends on the liquid to be handled. For acids or liquors of that nature, it is best to use wood, although it is not durable, yet it will last better than any form of metal; for salt, petroleum products, and strong alkalies, gun metal is the best; for all ordinary uses, however, cast iron is good enough, and if used with composition discs there will be no trouble from rusting fast even if the pump is idle for considerable periods.

With respect to valves for large pumping engines, they are best made of double beat type, and usually of the "Cornish" form. For this a design used by Mr. A. F. Nagle has proven very satisfactory; it is shown in Fig. 26. The features embodied in this design are a small width of beat to avoid any possibility of sticking and to get rid of surface on which the pressure of the water cannot act to open the valve; a diameter large in comparison with the lift so that the velocity with which the water comes to the opening shall be small; a shape which avoids all sharp curves and changes of direction and all possibility of air pockets. The valve has been found to embody all these qualities, to work noiselessly even at high speeds, and to follow the motion of the piston almost exactly.

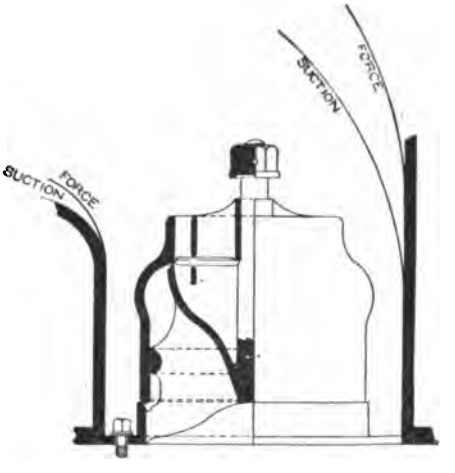

Fig. 26.

DESIGN AND CONSTRUCTION.

The essential parts of a reciprocating pump are inlet valves, a cylinder, a piston or bucket and outlet valves. The purpose of each is obvious. However, in order to get the best results certain points must be considered in the arrangement and construction of these parts. Some of these have already been mentioned in the

description of valves. Admission and outlet valves are usually of the same construction but sometimes differ in size, the outlet valves being smaller and more numerous so that the lift may be less and the slip under high pressures reduced. Valve decks are usually arranged, one for the inlet and one for the outlet valves; into these decks are screwed or bolted the valve seats, as already shown in Figs. 20—24, the valves extending above into a valve chamber as shown in Fig. 16. The valves should have space enough between them and between the outside valves and the walls of the valve chamber so that the area for the passage of the water shall equal that through the valve openings. Generally, considerations of strength of the valve deck also demand as wide spacing as does the necessary water way.

The design of the **cylinder** depends entirely on the kind of pump; the considerations involved having already been mentioned. For a small and compact machine, the piston type is the best and this calls for a smoothly-bored barrel. In order that this may be easily renewed, it is usual to make the bored portion as a lining which fits an outer casting so that when worn by grit or long use, the lining may be renewed without remaking the whole water end. Such a construction is shown in Fig. 16.

If there is much sand or dirt in the water, it is usually better to use some form of plunger pump, as the round-turned plunger is much cheaper to renew than the bored-barrel lining; also the cylinder will be cheaper as there is little boring to be performed. The difficulty of an inside packed plunger has been mentioned and also the alternative of a double plunger, outside packed. (See Figs. 17 and 18). In spite of the difficulty of ascertaining and stopping leakage past the piston, and the expense of boring and reboring or relining the cylinder, the piston pump is used, except for high pressure, more than any other form on account of its compactness.

The bucket is rarely used except in vertical pumps and then almost entirely for mine or well purposes. It is of the form of Fig. 25 with possibly the clack or hinged valve in place of the ball. The outside of the bucket is packed by cup leathers or by hemp packing, the same as a piston, or it may simply have grooves turned in the outside to form a water packing. The

barrel is often simply a brass pipe, bored to make a fit, and screwed to the end of the vertical pipe so that it may be easily renewed. In the case of deep wells, the bucket should be made with a long bearing surface so that it may the more easily keep a water-tight joint and may act as a steadier to keep the pump rod in line.

The **piston rod** for a pump is made much larger than considerations of strength would seem to demand, in order that there may be no vibration or deflection, as these would make it almost impossible to keep the stuffing box from leaking. Theoretically the rod for a long-stroke pump should be larger than that for a short-stroke of the same diameter; but, actually, the margin of strength is so great that the same size rod will answer for both. Allowance must be made in all cases for the reduction in diameter necessary to fasten the piston or plunger to the rod. This may be done by using a tapered end fitting a tapered hole in the body of the piston or by turning down the end of the rod and letting the piston bear against a shoulder; the latter method is the better. Usually a nut holds the piston in place, the nut being in turn secured by a lock nut or split pin through the end of the rod. Various modifications of these methods are of course used according to the fancy of the designer.

The diameters of piston and corresponding rod for a rod of cold rolled steel, tensile strength 65,000 pounds, a factor of safety of 10 and a pressure of 150 pounds are given in Table IV.

TABLE IV.

Piston diam. Inches.	Rod diam. Inches.	Diam. of rod end, Inches.
4	$1\frac{3}{8}$	$1\frac{1}{8}$
6	$1\frac{5}{8}$	$1\frac{3}{8}$
8	$1\frac{7}{8}$	$1\frac{1}{4}$
10	$2\frac{1}{4}$	$1\frac{3}{4}$
12	$2\frac{3}{8}$	$1\frac{5}{4}$
14	$2\frac{1}{2}$	2

The **water end** of the pump is necessarily long, as the piston must be of such length that it can be packed, and usually with some kind of soft packing such as flax, hemp or leather. Fig. 27 shows the detail of a piston designed for use with square flax packing; the body is fastened to the piston rod by the nut A, the packing laid in place, and the follower forced up by the nut B which is, in turn, secured in place by the lock nut C. For large sizes, the

design is the same except that the follower is set up by a number of nuts near the edge screwed onto stud bolts.

The leather-packed piston is made as shown in Fig. 28. The packing is made by pressing sheets of wet leather into iron forms and allowing them to dry there, after which the edges are finished, The radius of the corner must be large, so that the leather will not be injured in the pressing process, about $\frac{1}{2}$ to $\frac{3}{4}$ inch being sufficient. The diameter of the disc from which the cup is formed need not be more than $1\frac{1}{4}$ to $1\frac{5}{8}$ inch greater than that of the piston, as there is no object in having the lip of the cup longer than just sufficient to make a tight joint.

Less common forms of piston packing are the metal ring sprung into place and the series of grooves. For the former, the

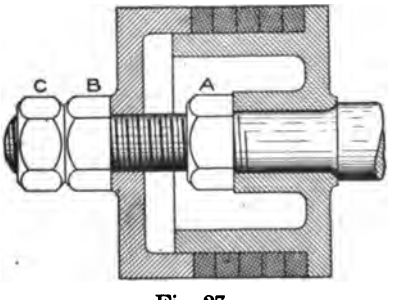

Fig. 27.

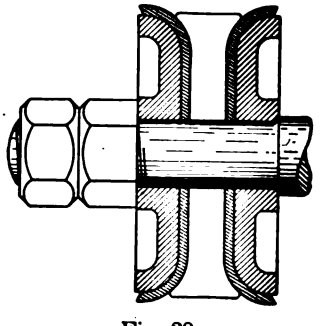

Fig. 28.

ring should be made about $\frac{1}{8}$ inch larger than the piston for a 12-inch piston; other sizes in proportion. For the groove packing, the grooves are best made about $\frac{3}{8}$ inch deep by $\frac{1}{4}$ inch wide. If soft packing is used, the length of a piece cut for a ring should be somewhat less than that needed to have the ends meet when placed around the piston body in order to allow for the swelling of the packing as it becomes wet. One rule is to make the space between ends equal to the width of the packing.

For a plunger, a hollow casting is generally used; the only requirement being that the thickness shall be sufficient to withstand the pressure against which the pump acts. The length must, of course, be equal to the stroke plus the length of the stuffing box and an amount sufficient to ensure that the head of the plunger will never strike the gland. A plunger is sometimes used with an

open end and the partition set back into the interior of the plunger, but this is not as strong a construction as the closed-end type and is more expensive to make.

With single-cylinder pumps it is customary to use an air chamber on the discharge to steady the flow of the water and prevent shock, but for those having several cylinders this is not considered necessary. Even if the motion is slow, and the pressure against which the pump works is great, it is better to use the air chamber even with multiple cylinders, and it is often advisable for fast running pumps or those with a long suction pipe to use an air chamber on the suction side as well. The volume of the discharge chamber should be three to four times that of the piston or plunger displacement per stroke, and for the suction chamber twice such displacement if the pump is single. For duplex, $\frac{1}{3}$ to $\frac{1}{2}$ less will answer. The neck of the chamber should be long and narrow so that the air will not easily escape, and the flow of water into and out of the chamber will be somewhat retarded. For small sizes the chamber is made of copper in a standard form as seen in Fig. 29, the dimensions being about as follows:

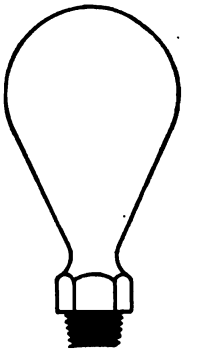

Fig. 29.

Diam. inches.	Height, inches.	Pipe-tap thread, inches.
6	10	1
8	14	$1\frac{1}{4}$
9	15	$1\frac{1}{2}$
10	16	2

For larger sizes a casting is used with straight sides and a hemispherical top as seen on Fig. 17. For this style a common proportion is to make the height three times the diameter and the diameter of the neck one-third that of the chamber proper.

The air chamber on the discharge should be placed at the highest point of the valve chest and above the delivery opening so that the air will not tend to slip out with the water. Even then, the air will be gradually absorbed by the water, and provision must be made for renewing the air cushion either when the pump is not in use by admitting air through a cock and allowing some water to escape from the valve chest, or continuously by an automatic pump.

The suction air chamber should be so placed that the stream of water flowing to the pump may cushion against the air in it without changing its direction abruptly. Two positions which fulfill this condition are shown in Fig. 30. If it is impossible to place the suction chamber in such a position, the capacity should be increased considerably, the amount depending on the speed at which the pump is to be run.

In designing the valve chest, flat surfaces should be avoided, as they are not adapted to resist pressure. A rounded or oval surface gives greater strength and will more easily conform to the outline presented by the valve studs and the top of the chest. A good design is shown in Fig. 16.

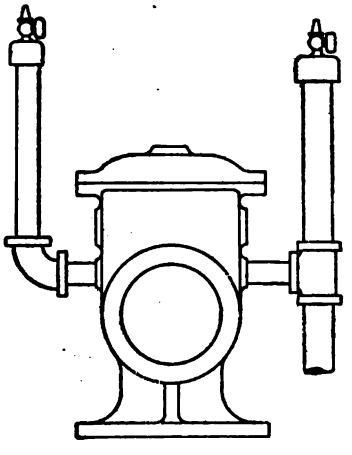

Fig. 30.

Clearance is the name given to the space at the end of the cylinder into which the piston or plunger does not travel. It is not, as is the case with the steam engine, a source of loss, hence it is advisable to provide a generous amount so that there will be a place for grit or foreign matter to settle, and no possible chance of the piston striking the cylinder head.

The ports or valve openings and the passages from the valve chests to the cylinder should be large enough so that the velocity of the water when the pump is working at its greatest speed will be not over 300 feet per minute. The ports should be short and direct to avoid friction, and if possible should be so arranged that the water passes into, through and away from the pump without changing the direction of its motion. This last condition usually conflicts with the necessity of placing the valves so that they are readily accessible and is, therefore, disregarded by many makers.

The stuffing-box for rods 1 inch or less in diameter is made with a cap to screw over the end of the box and forces in the gland as in Fig. 31. For larger sizes the gland is forced in by nuts screwed on stud bolts which pass through a flange on the gland. See **Fig. 32.** The gland and box may be bushed with brass as shown,

or made of solid iron. The box should be made deep enough to allow four rings of packing and should preferably have the bottom and the end of the gland chamfered in order to force the packing against the rod and make the joint more secure.

The frame or body of the pump is to hold all parts together and keep them rigidly in position. With this end in view, it pays to use enough iron to make the frame massive, yet it should be so distributed as to be of the most use. The standards or legs should be of good length so that the pump may stand well away from the foundation and be accessible. The feet should have a large sur-

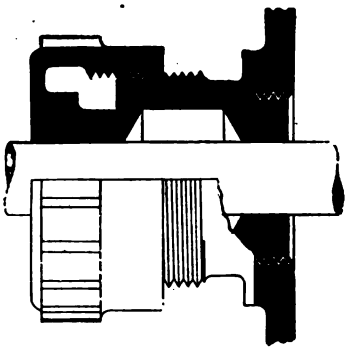

Fig. 31.

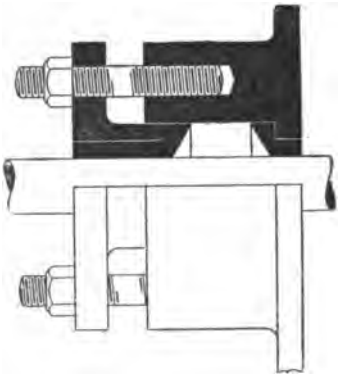

Fig. 32.

face to secure stability and good bearing. For pumps which are to work against high pressures, round chambers connected by round passageways should be used for the valve chambers. The valve chambers should be so arranged that the valves or seats may be easily removed and replaced, a convenient arrangement being that of Fig. 16 where the inlet valves are directly under the outlet valves, hence can be readily examined Details of such an arrangement as used in the Cameron and Davidson pumps are seen in Figs. 33 and 34 respectively.

The arrangement of the parts of a pump is the test of a designer's skill. Air pockets must be avoided in the cylinder or clearance spaces, as otherwise the pump will waste much of its effort on the air thus caught. The passage through the cylinder must be so designed that the water leaves at the highest point of the chamber and all recesses in which air might be caught must

be carefully avoided. Air pockets in the discharge valve chamber are not serious, but in the inlet valve chamber they will result in lost motion.

Almost all small horizontal pumps are arranged with both the inlet and discharge valve chambers above the cylinder, as this makes all valves readily accessible, and gives a compact water end. Unquestionably smoother action and a more efficient pump would be obtained if the admission valves were placed below the cylinder so that the water need not reverse its direction of flow while passing through the pump; but it is a case for judicious compromise, and the common arrangement gives good service so long as the speed of working is slow. Nevertheless, it is well to keep the better arrangement in mind, in which the suction enters at the bottom of the pump and the discharge leaves at the top.

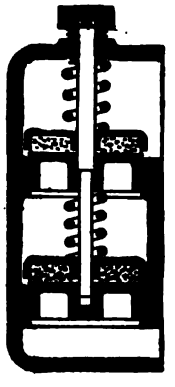

Fig. 33.

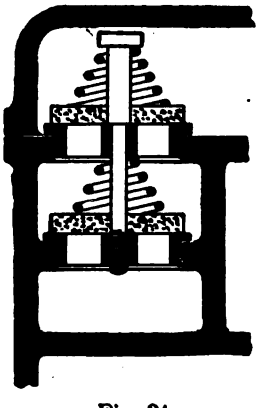

Fig. 34.

The suction pipe usually comes in at one side, and the discharge leaves directly above it on the same side, as shown in Fig. 35, or the side opposite. Fig. 35 is the cross section of a single cylinder pump.

For the direct-acting steam pump, the steam and water cylinders must have their axes coincident. The two cylinders should be placed as near each other as is consistent with provision for the valve gear and room to care for the stuffing boxes. Compactness is desirable, but convenience in caring for the pump and in making repairs is much more essential.

At the lowest point on each end of each cylinder provide a drain cock by means of which all water may be drawn off when the pump is shut down in order to prevent rusting or pitting and to avoid danger of freezing in cold weather.

The size of the piston or plunger and length of stroke are determined by the capacity desired. The allowable speed of piston is often stated as 100 feet per minute, but it is better

limited to 60 double strokes per minute, as the determining factor is really the time needed for reversal and changes of valves at the end of the stroke. For small pumps the limit might be made 100 double strokes per minute, but the smaller number is better. At 60 double strokes or 120 single strokes per minute the piston speed in feet per minute will be:

$$\frac{l \times 120}{12} = 10\,l$$

where l is the stroke in inches; and for 100 double strokes the speed will be $\frac{1}{6}$ times the above value or 16.7 l. For pumps of over 6-inch stroke this higher limit is not advisable.

Table V gives the capacities of various sizes of pump cylinders at 60 double strokes per minute, without allowance for slip.

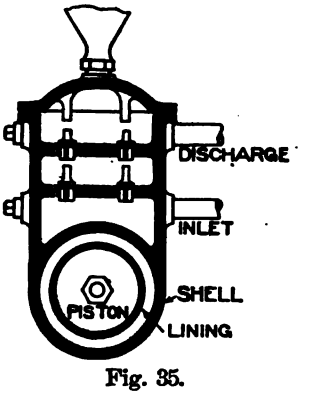

Fig. 35.

From Table V can be seen directly the capacity of any size pump at a speed of 60 strokes per minute, or the diameter and corresponding length of stroke needed to move any given number of gallons per minute. To find the capacity at any other speed, divide the given capacity by 60 and multiply by the number of double strokes at which the pump is to run. Thus to find the capacity of a pump 3 inches diameter by 4 inches stroke at 80 strokes; the capacity at 60 strokes is 14.7 and at 80 it will be

$$\frac{14.7}{60} \times 80 = 19.6 \text{ gallons per minute.}$$

The ratio of stroke to diameter may be chosen by the designer, the value in standard designs running from 2:1 to 1:1.25, the more common being 3: 2 and 1: 1.

The first value to be found is the amount of water to be handled per minute, and this will usually be determined by the work for which the pump is to be used. The allowance for "slip," that is, the loss due to slow closing of the valves and leakage must then be added before the size of the water end can be settled.

The slip depends on the style of valve used, the size of pump

TABLE V.

CAPACITY OF PUMPS IN GALLONS PER MINUTE AT 60 DOUBLE STROKES PER MINUTE.

Diameter	STROKE.														
	2	3	4	5	6	7	8	9	10	12	14	16	18	20	24
1	.0.81	1.21	1.63	2.02	2.42	2.85	3.26	3.68	4.10	4.90	5.70	6.52	7.36	8.20	9.80
2	3.26	4.89	6.52	8.15	9.78	11.41	13.04	14.67	16.30	19.56	22.82	26.08	29.34	32.60	39.12
2½	5.06	7.63	10.18	12.72	15.27	17.81	20.36	22.90	25.45	39.54	35.63	40.72	45.81	50.90	61.08
3	7.34	11.01	14.68	18.35	22.02	25.69	29.36	33.03	36.70	44.04	51.38	58.72	66.06	73.40	88.08
3½	9.95	15.00	20.00	25.0	30.00	35.00	40.00	45.	50.	60.	70	80	90	100	120
4	13.1	19.6	26.2	32.6	39.2	45.8	52.2	58.9	65.3	78.4	91.6	104.4	117.8	130.6	156.8
4½	16.6	24.8	33.0	41.3	49.5	57.8	66.1	74.4	82.5	99.0	115.6	132.2	148.8	165.0	193.0
5	22.0	30.6	40.8	50.9	61.1	71.3	81.5	91.6	101.8	122.2	142.6	163.0	183.2	203.6	244.4
6	29.4	44.0	58.8	73.4	88.0	103.	117.6	132.3	146.8	176.0	206.	235.0	264.6	203.6	352.0
7	39.9	60.	79.8	100.	120.	140	160	180	200	240	280	320	360	400	480
8	52.2	78.4	104.4	130.8	156.8	183.	208.8	234.8	261.6	313.6	366.	417.6	469.6	523.2	627.2
9	66.1	99.2	132.2	165.	198.4	237.	264.	297.	330.	396.8	474	528	594	660	793
10	81.5	122.4	163.	204.	244.8	286.	326.	367.	408.	489.	572	652	734	816	978
12	117.6	176	235.2	293.6	352.	412.	470.	529.	587.	704.	824	940	1058	1174	1408
14	160.	240	320.	400.	480.	560.	640.	720.	800.	960.	1120	1280	1440	1600	1920
16	209	313	418	523	626	732	836	939	1046	1252	1464	1672	1878	2092	2504
18	264	397	528	660	794	948	1056	1188	1320	1588	1896	2112	2376	2640	3176
20	326	489	652	816	978	1144	1304	1468	1632	1956	2288	2608	2936	3264	3912
22	395	591	790	988	1180	1380	1580	1780	1978	2360	2760	3160	3560	3956	2720
24	470	704	940	1174	1408	1648	1880	2116	2348	2816	3296	3760	4232	4696	4832

and the speed at which it runs. Approximate values are shown in table VI.

TABLE VI.

Class of Pump.	Slip in Percent of Volume Pumped.	
	At high speed.	At low speed.
Small Centrifugal	75	60
Medium "	50	40
Fire Engines	40	30
Large Centrifugal	25	15
Lift and Drainage	15	10
Mine and Deep Well Pumps	8	5
Waterworks	6	4

We can now determine the size of a pump to move a given volume of water. Suppose, for example, that it is required to force 200,000 pounds of water through a condenser per hour. This, at 8.338 pounds per gallon, would be 23,987 gallons per hour, or 400 per minute, if there were no slip; but for a common lift pump we must allow 10 per cent so that the capacity must be for $400 \times 1.10 = 440$ gallons per minute. From table V this would require a 10×12, a 9×14 or an 8×18; the 10×12 would be the most common proportion.

The area of valves should be such that the velocity of the water through them will not exceed 250 feet per minute; for 440 gallons, the volume would be $\frac{440 \times 231}{1728} = 59.8$ cubic feet. At 250 feet per minute, the valve area $= \frac{58.8}{250} = .2352$ square feet, or practically 34 square inches. The area of 3-inch valve is 7 square inches, so that the five valves would be sufficient. The circumference is 9.42 inches; this gives a lift of $\frac{34}{5 \times 9.4} = .723$ inch, an amount perhaps not too great for smooth action at the low pressure of a condenser pump; but it would be better even for this service to use twelve 2-inch valves giving a lift of .45 inch.

To find the power required, it is necessary to know the pressure against which the pump is to act both for suction and forcing. Assume this to be, in the present case, 5 feet of

suction head and 5 pounds pressure due to the friction in piping and condenser tubing; the total pressure will then be 7.16 pounds per square inch, and on the 10-inch piston the total force would be $78.54 \times 7.16 = 562$ pounds. The pump is to make 120 strokes per minute, each stroke being 12 inches or one foot long, so that the work per minute would be $120 \times 562 = 67{,}440$ foot pounds, and the power $67{,}440 \div 33{,}000 = 2.04$ horsepower provided there were no friction. It is customary to allow 25 per cent for friction in pumps, which would increase the power required to $2.04 \times 1.25 = 2.55$ horsepower.

To find the diameter of steam piston needed, the steam pressure must be known and the foregoing process can then be reversed. Assume the pressure to be 80 pounds; the work per minute is $2.55 \times 33{,}000 = 84{,}000$ foot pounds, which, at 120 feet per minute would require a force of 700 pounds, and, at 80 pounds per square inch an area of $\frac{700}{80} = 8.75$ square inches, corresponding to about a $3\frac{3}{8}$-inch diameter. Usually the steam and water ends are made much more nearly the same size than this would indicate and the steam throttled between the boiler and the steam end of the pump, so that the pump may be adaptable to a wide range of conditions.

The parts of a pump should be proportioned to withstand the forces which are to come on them, not only without danger of breaking, but without bending. For the parts which are to be of cast iron, such as cylinder, valve-chests and frame, the consideration which decides the dimensions is often the thickness of metal which will cast well; while for the piston it is the length which will give a good bearing surface on the cylinder to resist wear, and a tight joint to prevent leakage. These points are largely matters of experience and good judgment, but some ideas can be given as to common practice.

For a packed piston, usually 4 rings of $\frac{3}{4}$-inch square packing are used. Common values of rod diameters are given in Table IV. In designing the body and follower of the piston, it should be remembered that the less weight the better so long as the strength is sufficient; but a piece of cast iron less than $\frac{3}{8}$-inch thick seldom casts well in any complicated casting.

For the cylinder, the same equation may well be used for both steam and water ends. Whitham gives, allowing for strength, rigidity and possible reboring: $t = .03\sqrt{P\ D}$ where P is the maximum pressure in the cylinder in pounds per square inch and D the diameter in inches; t is of course the thickness of the cylinder walls. The length of the cylinder will be the length of stroke plus the thickness of piston and fastening nuts plus twice the clearance at one end which should be at least $\frac{1}{2}$ inch. The clearance should be counterbored to leave the length of the bore somewhat less than the length of stroke plus the length of piston. The soft packing or piston rings should not, however, be allowed to run into the counterbore. An average of various formulas for the thickness of cylinder heads and flanges as given by Kent, is $t = .00036\ P\ D + .31$ inches. The flanges on the cylinder are made of the same thickness as those on the cylinder head.

For large cylinders it is usual to support the head by ribs running from the center to the circumference, though some designers consider this bad practice. The flange through which the bolts pass is made $\frac{1}{3}$ thicker than the thickness of heads given by the above formula. To avoid springing of the flanges, cylinder-head bolts should be so spaced that the distance on centers will be not over 4 to 5 times the thickness of the flanges. The diameter should be such that the stress in the bolts will be less than 5000 pounds per square inch. If N is the number of bolts and d their diameter,

$$N \times \frac{\pi d^2}{4} \times 5000 = \frac{\pi D^2}{4} P, \text{ and simplifying,}$$

$$d = D\sqrt{\frac{P}{5000\ N}}$$

The area of inlet and discharge ports and pipes should be at least equal to that of the valves.

The thickness of the walls of the valve chambers is usually a little greater than that of the cylinder walls. For high-pressure pumps it is best to give these chambers as nearly as possible a cylindrical form in order to secure the greatest strength for a given weight of material.

For the frame, it is impossible to give any rules or suggestions except that the student should get catalogues from the

prominent makers and study them to see what has been found satisfactory.

For proportioning the driving end of a pump, whether steam, belt or motor driven, the ordinary rules for the kind of machinery involved apply. However, it should be remembered that the parts are subjected to severe shock on the reversal of the pump, hence a factor of safety as high as 15 should be used.

Where pumps are required for special service, the amount of water to be pumped per hour is generally fixed by the conditions of the service; but for a few cases the method of determining that amount can be definitely given. For a **boiler-feed pump**, it is necessary to know the horse-power of the boilers to be cared for by the pump. One boiler horse-power will call for the handling of 30 pounds of water per hour, and the pump should be specified accordingly with an allowance for good measure. For an air pump to remove the air and condensed steam the amount of fluid to be handled is difficult to estimate, hence, a common rule is to make the volume of the double-acting air-pump cylinder $\frac{1}{20}$ that of the low-pressure cylinder of the engine which it is to serve. The double-acting circulating pump has a volume about $\frac{1}{35}$ that of the low-pressure cylinder, or it may be figured from the horse-power of the engine, assuming the steam consumption at 25 pounds per I. H. P. per hour, and the cooling water at 30 pounds per pound of steam. Of course for accurate calculation, the steam consumption would vary with the size and type of engine, but the above will give safe average values.

Fire pumps are designed with special care to meet the requirements of the insurance companies. Stated briefly these are: Ability to start instantly after long disuse; must be duplex, with cross-operated valves; should preferably be of plunger pattern; must be brass-fitted throughout; designed to carry a water pressure of 320 pounds; suction valve area should be not less than 50 per cent of plunger area for 10-inch stroke and 56 per cent for 12-inch; discharge valve area should not be less than $\frac{3}{5}$ the suction-valve area; valve springs should be cylindrical, not conical, and held at the ends in a groove; studs to be so designed as to always allow a lift $\frac{1}{5}$ of the valve diameter; valve seats should be of gun metal held in the deck by a taper thread, or a smooth

taper bore forced in, having the lower edge turned over; least area of exhaust passage should be 4 per cent of piston area; admission ports should be not less than $2\frac{1}{2}$ per cent of piston area; clearances should be as small as possible; valve tappets should be non-adjustable; cushion valves controlling a by-pass from steam to exhaust port to regulate the amount of steam cushion at ends of stroke are recommended for 750 and 1000 gallon pumps; a gauge should be provided which will show at all times the length of stroke that the pump is making; water-pressure gauge with $\frac{1}{4}$-inch lever cock should be connected close to the air chamber; a relief valve of Ashton, Crosby or similar pattern and of size sufficient to discharge the full throw of the pump working at $\frac{2}{3}$ speed should be furnished and set to 100 pounds pressure; this valve should have a hand wheel conspicuously marked showing the direction to ← OPEN ; relief valve should discharge downward into a vertical pipe, thence into a funnel; $\frac{3}{8}$-inch brass drip cocks with lever handles should be on both ends of the steam and water cylinders; a $\frac{1}{4}$-inch lever air-cock should be on the cover over the water cylinders; each pump should be fitted with brass priming pipe, starting with a $2\times1\times1$ inch brass tee close to the pump beneath the delivery flange and leading to four $\frac{3}{4}$-inch valves, one opening to each of the four plunger chambers; a priming tank should be provided with bottom at least 5 feet above the pump, and having a capacity half that of the pump in gallons per minute, unless water flows to the pump under pressure; priming tank must be used only for that purpose; each pump is to be fitted for hose connections as per number of streams to be served; capacity of cylinders, sizes of piping, air and suction chambers and relief valves should be as per Table VII.

A special acceptance test is demanded as follows: The pump must run smoothly and without slamming at its full rated speed, maintaining a water pressure of 100 pounds per square inch when furnished with steam at a pressure of 45 pounds for the 500-gallon pump, 50 pounds for the 750-gallon pump, 55 pounds for the 1000-gallon pump. The water to be discharged through $1\frac{1}{8}$-inch nozzles on hose lines 150 feet long; the hose must lie quiet, showing uniform delivery; with all water outlets closed and steam supplied to give 80 pounds water pressure, leakage must not allow

TABLE VII.

Dimensions and Capacities of Fire Pumps.

No. of 1⅛-inch Streams.	Capacity, Gals. per min.	Ratio of Piston areas.	Boiler H. P. needed.	Steam press. needed at pump. Lbs. per sq. in.	Nearest commercial Size. Inches.		
					Diam. steam cylinder.	Diam. water cylinder.	Stroke.
One. (Too small except for auxiliary.)	320	4 to 1	....	40	10	5	10
					12	6	10
					10	5	12
					12	6	12
Two. (For small mills.)	500	4 to 1	100	40	16	8	10
					14	7	12
Three. (For general use.)	750	3 to 1	115	48	16	9¼	10
					16	9	12
Four. (For large factories.)	1,000	3 to 1	150	44	18½	10½	10
					18	10	12

For over 1,000 gallons per minute, two pumps should be used.

Capacity, Gals. per minute.	Diameter of pipes in inches.				Capacity, gallons.		Diameter Relief Valve, Inches
	Suction.	Discharge.	Steam.	Exhaust.	Air chamber.	Suct. cham.	
320	6	5	2½	3	10	8	2½
500	8	6	3	4	17	13	3
750	10	7	3½	4	25	18	3½
1,000	12	8	4	5	30	24	4

the pump to make more than one double stroke per minute; with water outlets nearly closed and pump running slowly it must carry water pressure of 240 pounds per square inch and all joints remain practically tight; with relief valve set at 100 pounds and all other outlets closed it must discharge full delivery at 50 double strokes per minute with the water pressure rising to not over 125 pounds.

These specifications call for a first-class design and construction, but such are necessary if a fire pump is to serve its purpose of a protection in case of emergency.

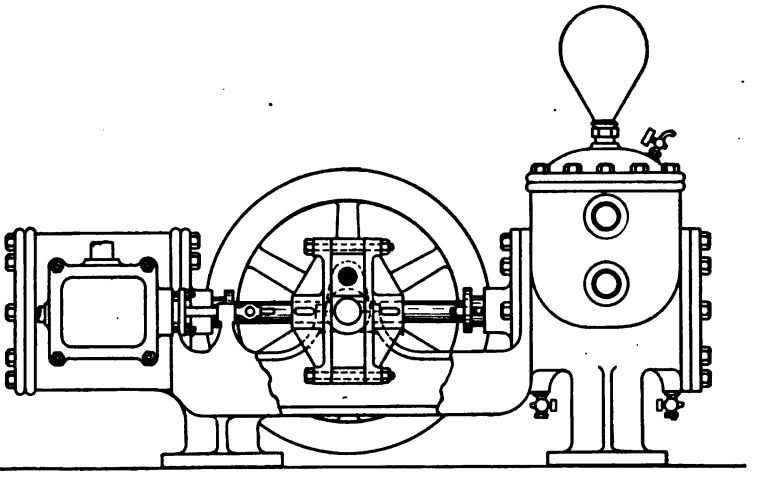

Fig. 36.

A name plate must be placed on the inboard side of the air chamber bearing data as follows in black enamel letters $\frac{1}{2}$ inch high on a white enamel ground:

Diameter of cylinders and stroke $16\times9\times12$. Capacity gallons per minute 750 or three $1\frac{1}{8}$-inch smooth nozzle streams. Full speed, revolutions or double strokes. "For Fire purposes never let steam get below 50 pounds, nights or Sundays."

TYPES.

In classifying pumps by types, two methods of division naturally suggest themselves; the first according to the arrangement of the water end, the second according to the means of driving employed. The difference between single-acting and

double-acting pumps has already been noted; the single, duplex or triplex arrangement, depends on whether one, two or three cylinders are assembled into a single machine, drawing water from the same suction inlet and discharging it into the same outlet.

The single pump is much used for small sizes or where con-

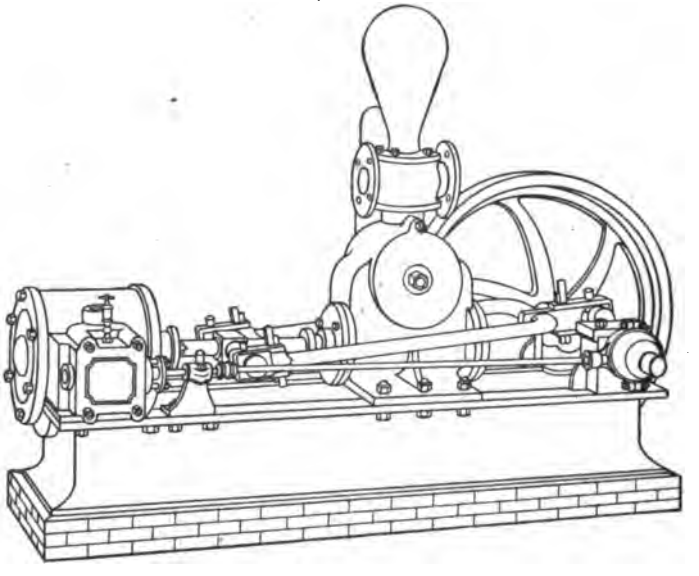

Fig. 37.

stancy of flow is of minor importance. It is, of course, the cheapest arrangement and gives satisfactory results if the pump is of ample size for the work. A difficulty with this type of pump is the provision of a means of operating the steam valve to reverse the motion of the piston at the end of the stroke. The pistons in steam and water cylinders must be stopped gradually to avoid pounding, and this leaves no force available just at the end of the stroke when it is needed to move the steam valve so as to reverse the motion.

The difficulty has been overcome in two ways; first by the introduction of a shaft and fly wheel, driven by a crank and a yoke in the rod between the steam and water cylinders as seen in Fig. 36. The inertia of the fly wheel then furnishes the force needed to move the valve at the instant of reversal, generally by

means of an eccentric on the shaft. This form is often used for fire-engines, two or three sets of cylinders sometimes being connected to one shaft, in which case the pump becomes of duplex or triplex form. For stationary pumps, the fly wheel and shaft are more commonly placed beyond the cylinders at one end of the frame, and the shaft is driven by a crank and connecting-rod mechanism from a crosshead attached to the piston rod as seen in Fig. 37. The fly wheel also makes it possible to use steam expansively. The second method of effecting reversal is by the use of an auxiliary valve and steam-driven main valve. This is the system generally found on small, single-cylinder, direct-acting pumps for boiler feeding, tank filling and similar uses. The details of the devices used vary greatly with different makes; these will be taken up later.

The single pump is compact, cheap and convenient, but the flow of water is by a series of impulses rather than in a steady stream, so that, if the pump is forced, it slams badly and produces water hammer, an evil but partially remedied by the use of discharge and suction air chambers.

Duplex pump. To avoid this, and at the same time to simplify the moving of the steam valve, the late Henry R. Worthington devised the duplex form, in which two direct-acting pumps are mounted side by side, the water ends and the steam ends working in parallel between inlet and exhaust pipes as seen in Fig. 38. The steam valve for cylinder A is moved by a bell-crank lever driven from the rod of cylinder B as it nears the end of its stroke; so that, when the piston of B is about to come to rest, that of A is set in motion. While the piston of A makes its stroke, that of B is at rest until that of A, near the end of its stroke, moves the steam valve of cylinder B by means of a second bell-crank lever, and the piston of B is once more set in motion. Thus the pistons move alternately, but one or the other is always in motion and the flow of water is made practically continuous. By this arrangement the auxiliary steam valve is made unnecessary and a simple D valve, driven by a valve rod running through a stuffing box, is used.

The triplex pump having three cylinders delivering to the same outlet is generally used for power pumps; the pistons or

plungers are driven from a shaft by three cranks or eccentrics, set at 120 degrees to each other so that the turning effort required on the shaft may be as uniform as possible and the flow of water steady. A style often used is single-acting with trunk plungers, that is, the connecting rod which drives the plunger is fastened to a pin inside the hollow body of the plunger as seen in Fig. 39. This gives a compact form and one not likely to get out of order, but the trunk arrangement is inconvenient in case of any wear on the pin inside the plunger, and the side thrust of the connecting rod creates a certain tendency to wear the plunger packing so that

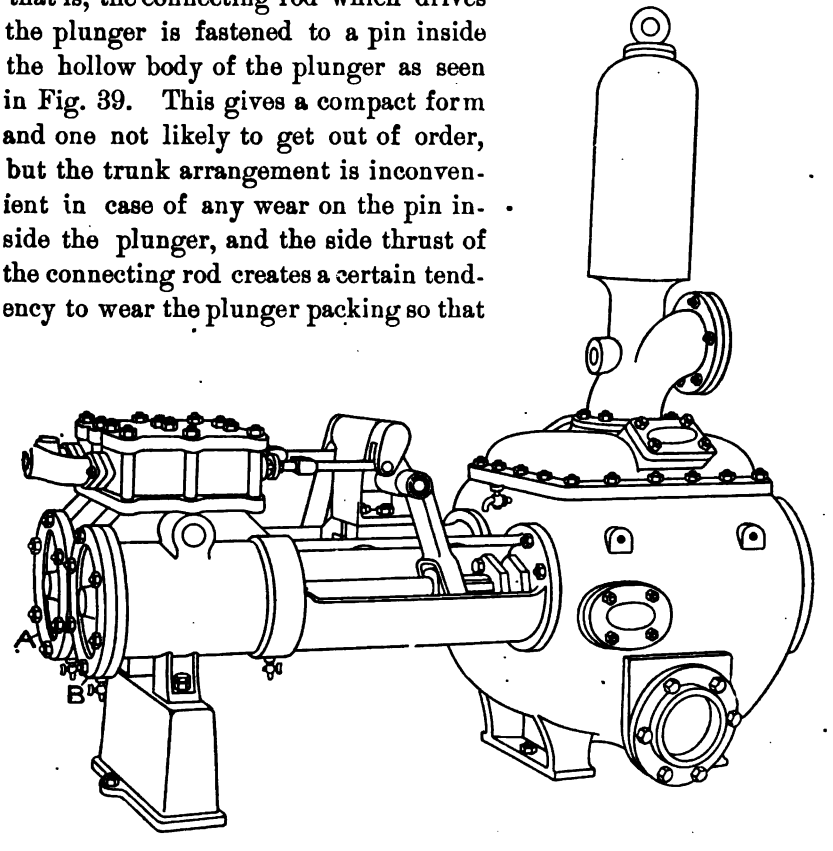

Fig. 38.

it will not remain tight. On the whole it is better, though more expensive, to have a regular crosshead and ways to receive the thrust of the connecting-rod as in Fig. 42. The cylinders in this style of pump are usually vertical and the crank shaft is driven through a pair of reducing gears from a second shaft which is, in turn, driven by a pulley and belt or an electric motor.

Methods of Driving. With respect to the method of driving, the most common type of pump is the direct-acting, steam-driven,

with steam piston and water piston or plunger on opposite ends of the same rod. For small sizes, the steam cylinder works without expansion of steam, the piston being driven by full boiler pressure throughout the stroke. In fact it is impossible to avoid this feature of full-stroke admission, since the pressure in the water end is constant throughout the stroke, unless some means, such as a fly wheel or accumulator, be provided, a complication which is undesirable with small powers. The simplest way to make any use of the expansive force of the steam is by compounding; that is, letting the steam exhaust from the small, high-pressure cylinder, into which it is first admitted, to a larger low-pressure cylinder, placed in tandem, with the first one. See Fig. 40. This arrangement gives a pressure on the low-pressure piston, which gradually decreases from the beginning to the end of the stroke while the pressure on the high-pressure piston is constant, so that whatever useful work is gained from expansion is in the low-pressure cylinder. This is, however, found to be sufficient to pay for the extra cost if the service is continuous, in sizes having a water cylinder over 8×12 inches. The compound pump is nearly always duplex, though a few manufacturers offer a single tandem-compound style. Since the steam pistons must move together, the steam valves must move together, and a single motion serves to operate the valves for both high and low-pressure cylinders.

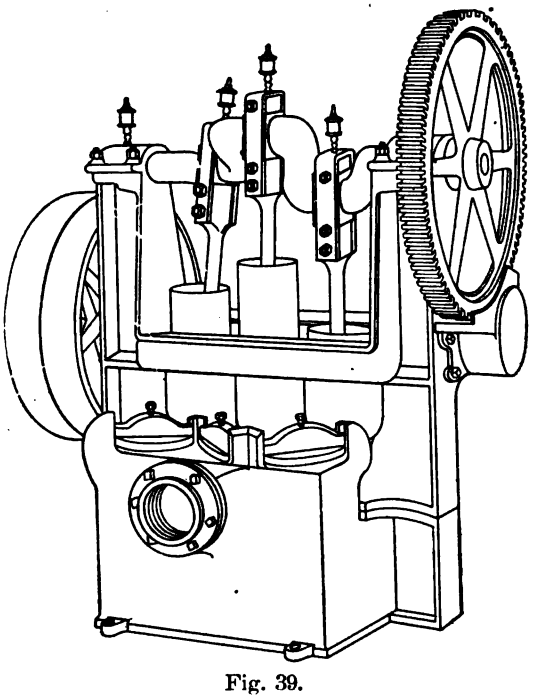

Fig. 39.

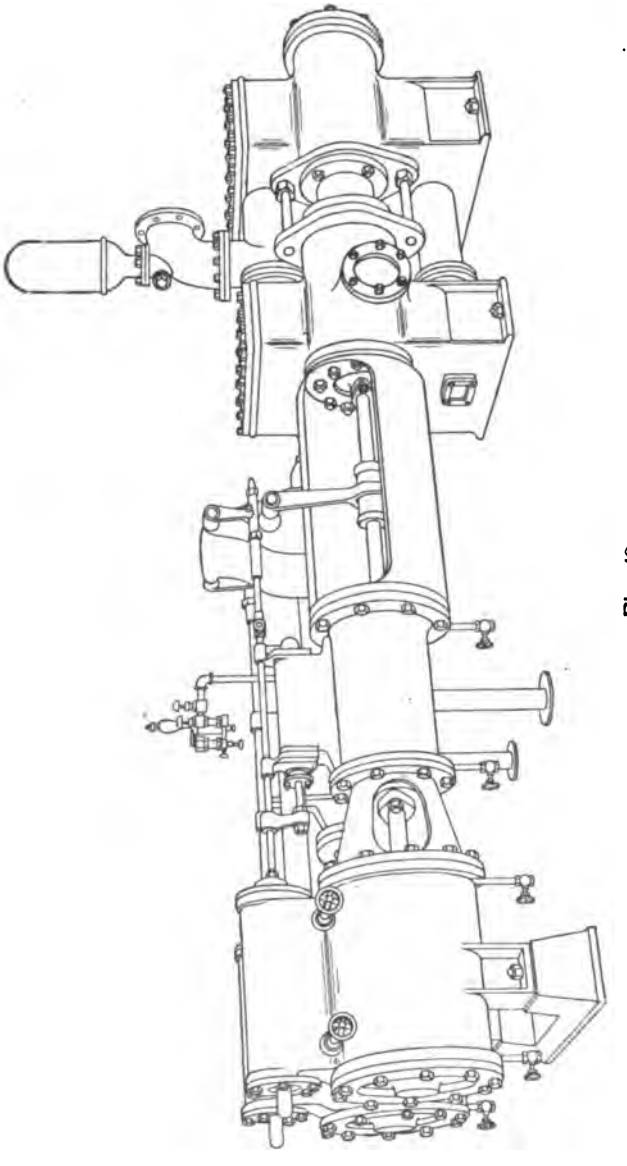

Fig. 40.

The simple steam-driven pump requires from 60 to 120 pounds of steam per horse-power, per hour; the gain by compounding is about 30 per cent per 100 pounds increase in boiler pressure, so that the steam consumption will be 40 to 95 pounds per horse-power hour.

The volume of the low-pressure cylinder is made from 2 to 4 times that of the high-pressure, depending on the initial pressure to be used.

The large steam consumption of direct-acting pumps of the single-cylinder type has led to the adoption, in many large stations, of power-driven pumps of various kinds. It is probably true that the cost for steam is less by this method when the power is developed by large engines, even after the losses in transmission are accounted for, because the efficiency of the engine is so much greater than that of a small pump. But, for small plants, the convenience of having the pump located near its work and of being able readily to control the speed has overbalanced the consideration of saving.

As previously mentioned, nearly all power pumps are geared, the power being applied to a driving shaft as seen in Fig. 39. The belted type is driven from a countershaft and provided with fast and loose pulleys used for starting and stopping, but there is usually no means of speed regulation. This, together with the expense of belting, the room needed and the fact that the convenient location for a pump is likely to be such as to make a belt a nuisance have prevented a general use of this type, and with the general introduction of electric power into all plants, the motor-driven pump is superseding it.

The motor-driven pump is of the same general form as the belted, but has an electric motor mounted on the driving shaft in place of the pulley, or has a double-reduction gearing as in Fig. 41. The high speed at which motors run make this double reduction necessary except in the case of large pumps. The only alternatives are the use of a large, slow-speed motor or a belted motor, either of which would be cumbersome and expensive. As to efficiency, probably there is not much difference between the belted and the motor-driven types. The electric method of transmitting power from the engine to the pump will be more econom-

ical than belting, but the transformation in the motor and the loss in the extra set of gears will offset this.

The speed of the motor can, however, be controlled by a starting box, or rheostat in the field circuit, hence the motor-driven has a considerable advantage over the belted type.

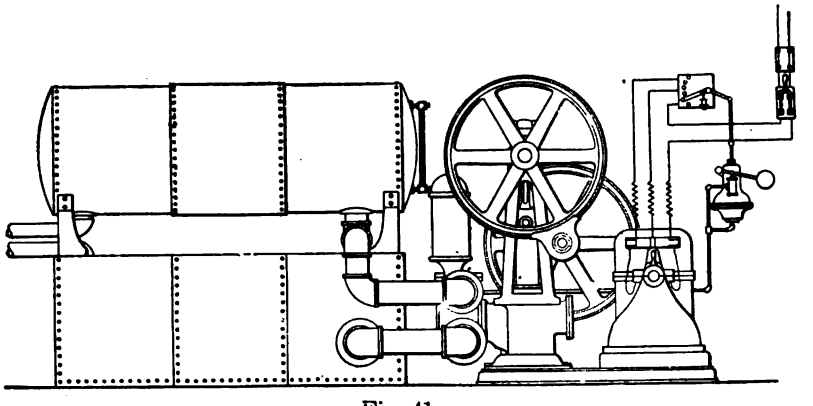

Fig. 41.

As compared with the direct-acting type the economy would be about like this:

Direct-acting—steam consumption, average, 90 pounds per hour per H. P.

Belt-driven—steam consumption of engine, per B. H. P. hour, average 35 pounds; at an efficiency of transmission for belting of 70 per cent and for gearing of 85 per cent, this would require about 59 pounds of steam per hour per H. P. delivered to the pump, and at an efficiency of 75 per cent for the pump this would give about 78 pounds of steam per hour per H. P. of effective work.

Motor-driven—steam consumption as before, 35 pounds; efficiency of dynamo 93 per cent, of transmission 90 per cent, of motor 93 per cent, of gearing 72 per cent, of pump 75 per cent, gives a steam consumption per effective H. P. of 83 pounds per hour.

Of course with large units the efficiencies may be better, but the relation would be about the same. The greater convenience of the direct-acting pump as to size and means of control, and the

ability to start it when the engine is not running have weighed in its favor in the majority of cases.

For some few places a gas or oil engine has been used for pump driving. In this type a clutch must be introduced between the engine and the pump, Fig. 42, as the engine will not start under load. The engine must first be started, then the pump thrown on empty and finally the valves operated so that the pump will begin to work. There is no means of varying the speed, but a regulator is sometimes introduced which throws out the clutch when the water reaches a certain height in the tank or reservoir

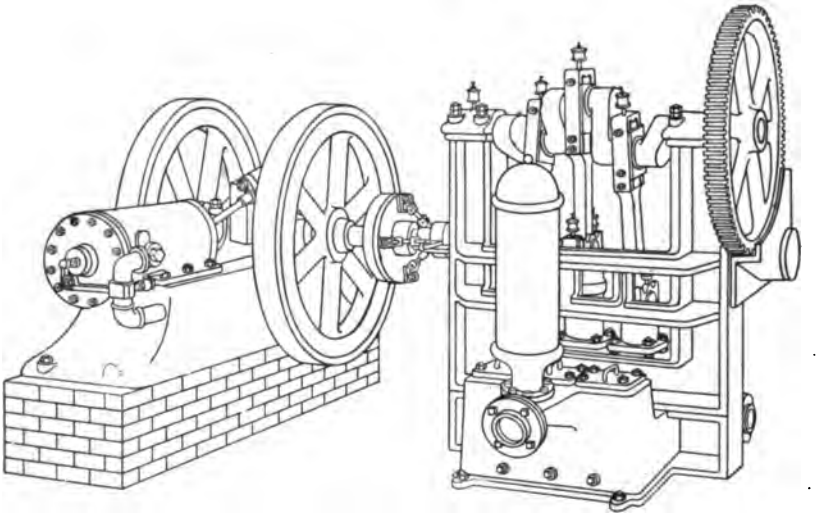

Fig. 42.

to be filled. This type is used only for such service as water supply for a house or reservoir system, and would hardly be desirable elsewhere.

To economize in floor space, the vertical marine type of pump is used. This is generally duplex, but occasionally single, and except for the vertical arrangement is identical with the horizontal type.

Another special type is the **combined-condenser pump,** Fig. 43, in which the steam cylinder is placed in the center and drives on one end of the piston rod a vacuum pump for drawing the condensed steam and air from the condenser, and on the other end a

pump for forcing the cooling water through the condenser tubes. The steam cylinder is the same as for any single pump except that there is a stuffing box at each end; the water end is the same as for any pump; but the air-pump end has composition disc

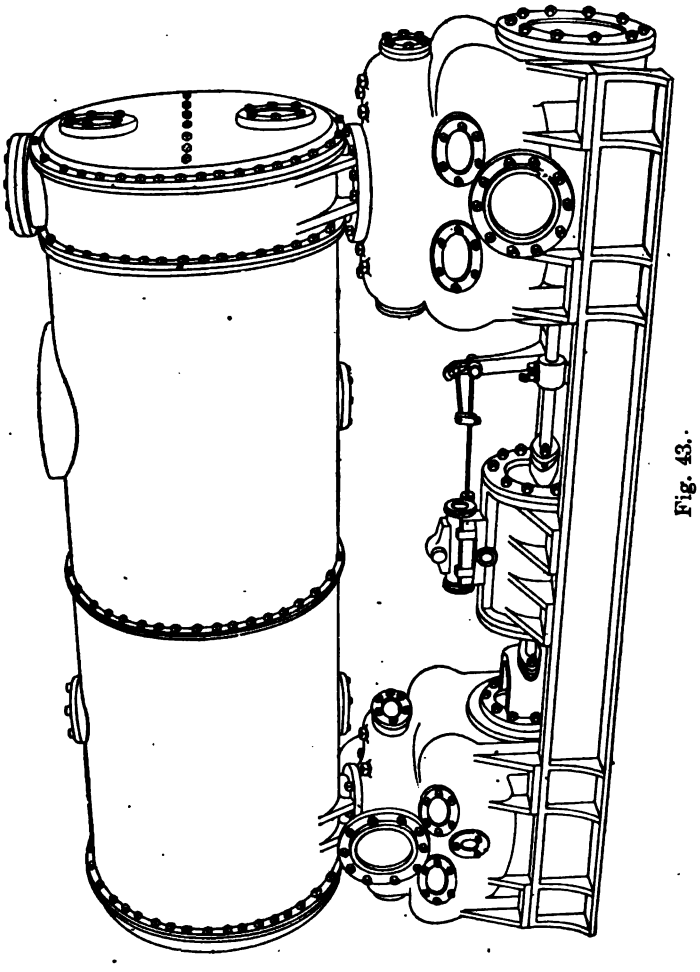

Fig. 43.

valves with an area of opening larger than that for a water cylinder of the same size, and the springs which close the valves are set at a less tension than for water.

Air pumps of special forms for condensers are often used combined with the condensers; they are driven by a separate engine or by a bell-crank lever from the crosshead of the main

engine. The cylinders are often vertical and single-acting on the lifting principle with valves in the top of the bucket. A unique and in some ways commendable style is that of Fig. 44 which avoids inlet valves and valves in the bucket, yet removes both air and the water of condensation quickly and effectively. Condensed steam and air flow into the lower recess A through the passage B from the condenser; as the piston descends, it forces the water from A so that it shoots up in the direction indicated by the dotted lines through the ports *p p* into the cylinder C. The piston rising, closes ports *p p* and catches the water and air in the cylinder forcing them out through the valves *v* in the top into the discharge chamber D, whence the mixture escapes through the exhaust passage K to the hot well, or wherever may be desired.

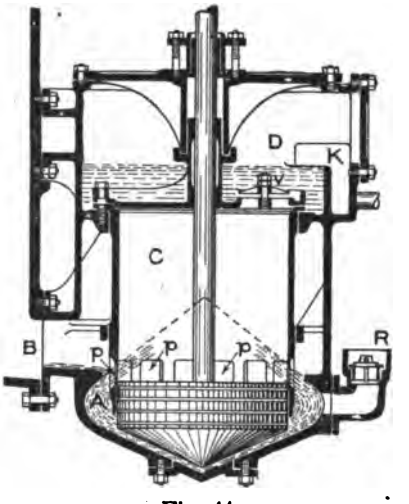

Fig. 44.

For certain uses, such as keeping up the pressure on a hydraulic elevator system, draining a pit or maintaining the water level in a tank, it is essential to have an automatic method of control which shall start and stop the pump as may be needed. For direct-driven steam pumps this is accomplished by control of a damper valve or a quick-acting gate placed in the steam pipe and for the motor-driven pump by a switch on the starting resistance box. In the one case, a float is moved with the water level, in the other a diaphragm worked by the tank pressures actuates a series of levers as seen in Fig. 41 so as to work the steam valve or switch as the case may be.

For high pressures and high speeds what is known as the Riedler System of mechanically operated valves (Fig. 45) is found especially desirable. The valve is of the poppet type, made so large that one valve answers for each end of a cylinder and is opened and closed by a rocking arm N, turned by a link and

driven from a wrist-plate much as in a Corliss-engine valve motion. The mechanical driving permits a high lift, with rapid opening and closing at the end of the stroke, thus decreasing the slip and shock. The high lift, 1 to 2 inches, and large water passages decrease the friction while the mechanical operation of the valves also allows of the high piston speed necessary to secure economy in steam consumption.

Referring to Fig. 45, the valve body P has on its lower face

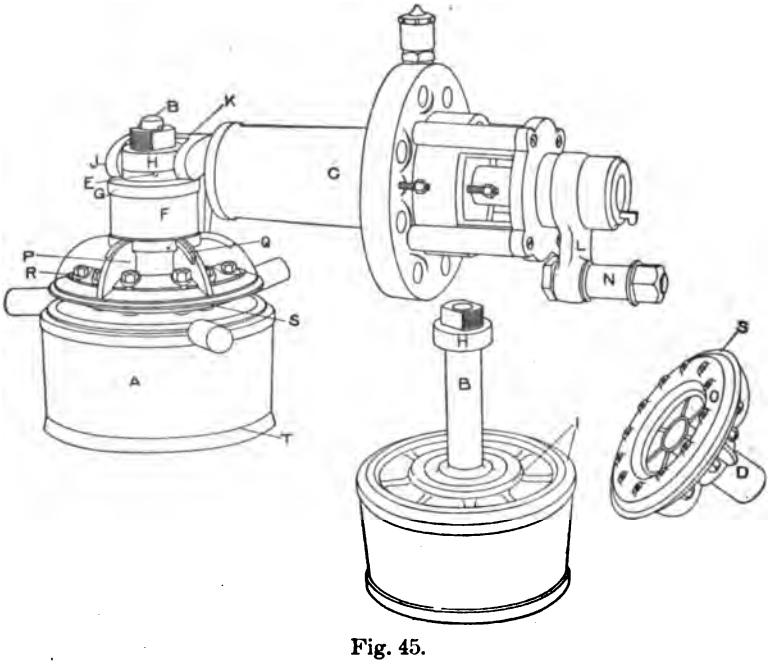

Fig. 45.

a leather seal S secured by a plate O. The sleeve D works on the spindle B, raised by a pin in the arms K which passes under the collar H and is lowered by the pressure of the blocks J on the cushion plate G. F is a rubber buffer to take the shock in case an obstruction gets into the valve. The arms K are mounted on and turned by a spindle which passes through the bushing C and is rotated by the crank L and crank pin N. One valve is used for inlet and one for outlet on each cylinder, as seen in the right-hand plunger of Fig. 46; for the left-hand end, the outlet valve of the larger plunger acts as an inlet and the clack valve between passages

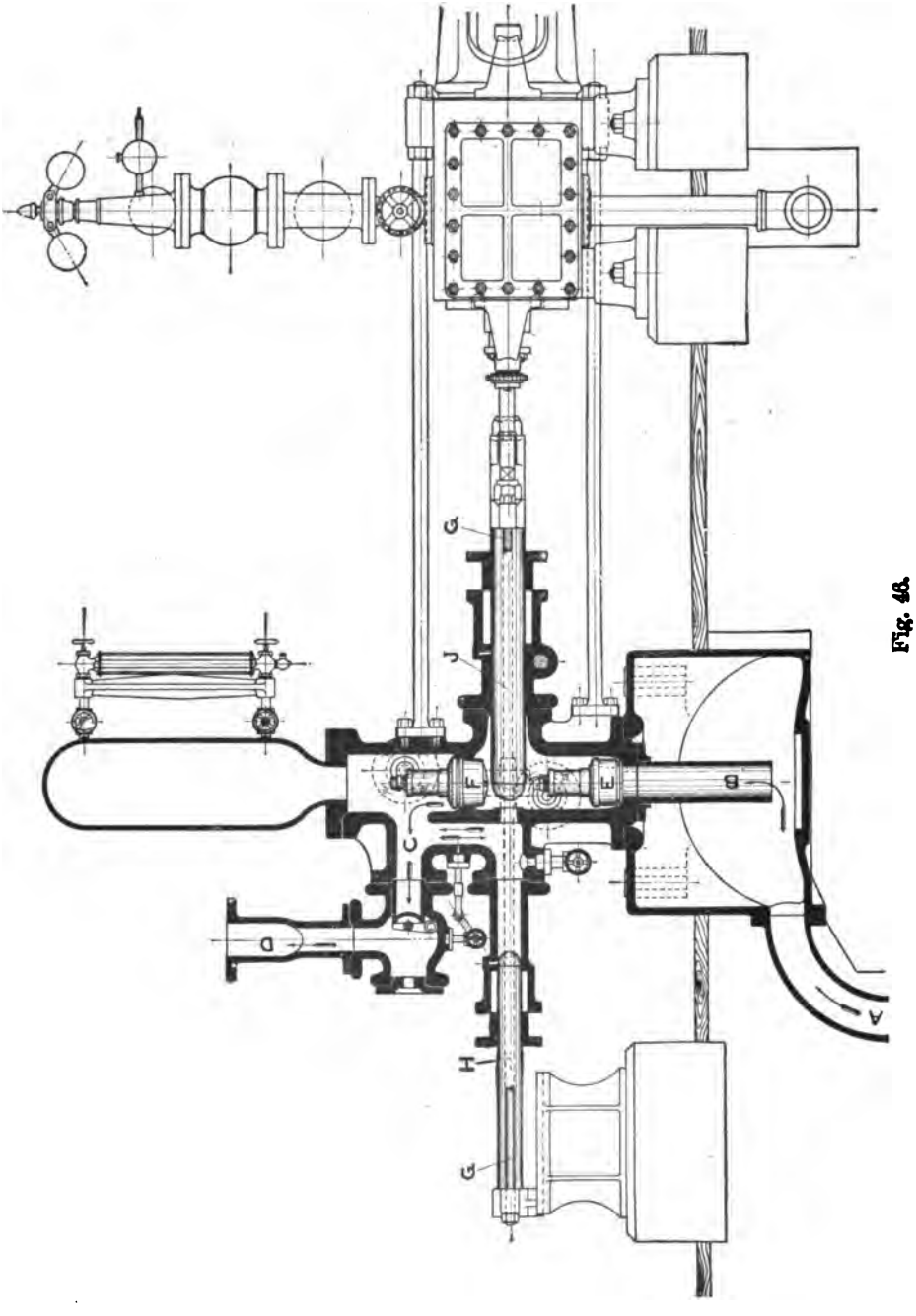

Fig. 46.

C and D acts as an outlet valve. The differential type of pump here shown is used for high pressures and to give a continuous flow with few valves; the cross-section of plunger H (Fig. 46), is half that of J, hence, on the stroke towards the left, half the water discharged by J is thrown into the small plunger chamber and half into the passage D; on the stroke towards the right, the half which entered the small plunger chamber is driven into chamber D and the large plunger draws its full quantity through the suction pipe B. The large passages and valves, the ample discharge and suction air-chambers, together with the positive opening of the valves, combine to make this pump very efficient and quiet in running. However, it requires a large space in proportion to its capacity while its complexity makes it expensive, so that it is available only for large sizes and in places where the space occupied is not of special importance.

STEAM VALVES.

The simplest form of valve is that used with a fly-wheel pump and which is moved by an eccentric as seen in Fig. 37; the detail is shown in Fig. 47 and the action as follows:

With the valve *v* in the position shown, steam is flowing through the port *a'* into the right-hand end of the cylinder and forcing the piston towards the left; at the same time, any steam which may be in the left-hand end of the cylinder is driven out through the port *a* into the exhaust passage *b* and thence to the exhaust pipe. If the shaft M be turning as indicated by the arrow, the eccentric *e* which is keyed to it, will also turn, and sliding in the eccentric strap S will draw it towards the right, and with it the valve. When the eccentric has made a fourth of a revolution it will be vertically above the shaft, and will have drawn the valve over so that it covers both ports, while, at the same time the piston will have moved so that it will stand at its extreme left-hand position. As the motion continues, the left-hand port *a*, will be opened to steam and the port *a'* to exhaust, causing the piston to reverse and move towards the right, which will continue until the eccentric has moved the valve to its extreme right-hand position and back to cover both ports. The eccentric standing directly below the shaft, the piston will again reverse and start towards the left, and the cycle be repeated.

This style of valve is called the D valve, and is much used for pumps and for the smaller reciprocating steam engines. For the ordinary pump the valve is made so that when in mid position it just covers the ports, and steam will enter one or the other end of the cylinder, if the valve is moved from that position. Steam will then be admitted full stroke and work without expansion in the cylinder. If it is desired to utilize some part of the expansive force of the steam, "lap" must be added to the valve as shown at *a*, Fig. 48, so that the port may be closed before the piston reaches

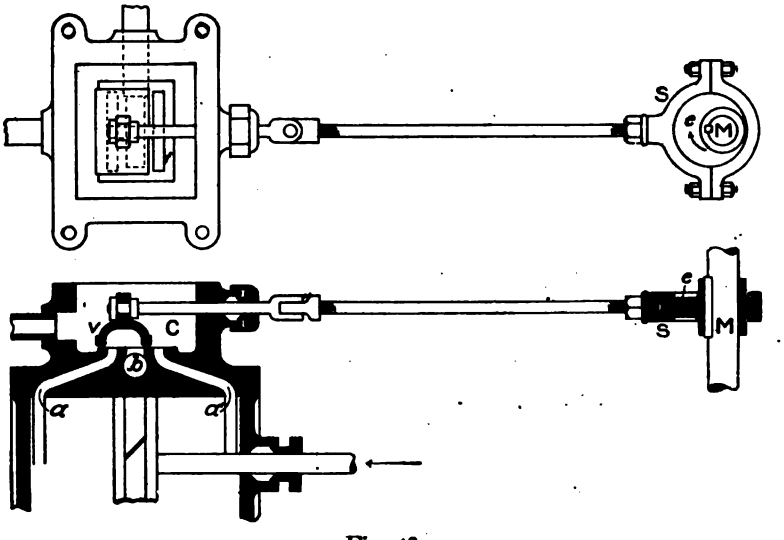

Fig. 47.

the end of the stroke; then the eccentric must be turned forward on the shaft in the direction of its motion in order that the valve may open by the time the piston is ready to reverse, and then "inside lap," *b*, must be adjusted so that the exhaust port will neither open too soon, thus wasting the expansive energy of the steam, nor close too early, thus causing too great a compression of the steam caught between the piston and the end of the cylinder.

The construction is the same as for a simple reciprocating engine, which indeed it is, with the addition of a pump cylinder and piston attached to the end of the piston rod.

Another form of valve common in pumps is the B type, Fig. 49. The action is the same as for the D type, except that steam

flows into the cylinder through the cup-shaped cavity in the valve instead of past the end, and the motion of the valve relative to that of the piston is the opposite of that for the D valve. The B type is adapted for use with the steam-driven valve to be described, but is not suited to be driven by an eccentric. The steam-driven valve is used on the great majority of small single pumps on account of the compactness, light weight and low cost and in spite of the poor steam economy. In a few of the first named pumps, for instance, the early Worthington, single type, the main steam valve was moved by tappets on the valve rod which were struck by an arm fastened to the piston rod. This was not certain in its working and has been displaced by the main valve moved by a small piston

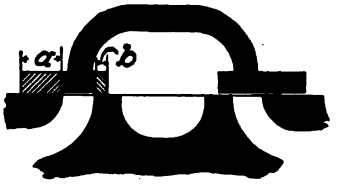

Fig. 48.

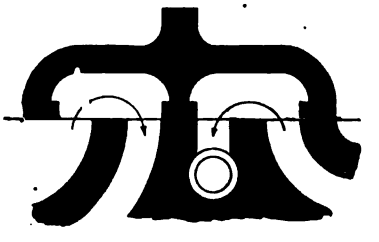

Fig. 49.

working in an auxiliary cylinder. Steam is admitted to this cylinder by an auxiliary valve moved by a tappet or lever mechanism worked from the main piston rod.

The designs of different makers vary more in this detail than in any other, and we shall now discuss a few typical examples at some length.

The first to be developed was the Knowles, Fig. 50. The chest piston A is moved by steam admitted through ports *m* and *z* between the end of the chest piston and its cylinder, and carries with it the main steam valve B, which opens and closes the ports to the main steam cylinder. The ports *x y z* and *x' y' z'*, of the small detail view, are made to register with the ports *m n* and *m' n'* by the turning of the chest piston by the tappet C driven by the rocker-bar R and the roller E. As the main piston reaches its left-hand position, the roll E will lift the left-hand end of the rocker bar and place the ports so that steam is admitted to the right of the chest piston and exhausted from the left; the main valve will then be carried to the position shown in Fig. 50 and

steam be admitted to the left-hand end of the main steam cylinder in order to drive the main piston towards the right. At the other end of the stroke the operation will be reversed. The arm F will not strike tappets C or D unless the steam fails to move the main valve, in which case the piston A would be driven by pressure on the tappets, but with more or less slamming depending on the speed.

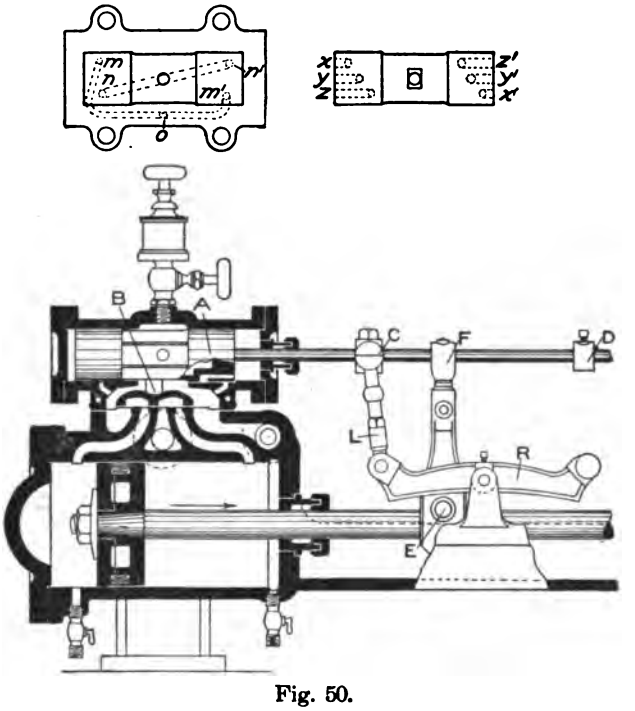

Fig. 50.

The third set of ports in the chest piston and its seat are to form a connection between the two ends of the auxiliary cylinder after the chest piston has moved a certain distance in order to prevent its striking the end of the chest. The length of stroke can be changed by altering the height of the roller E, and unevenness of stroke at the ends can be remedied by changing the length of the link L.

An entirely different style of valve is that of the Cameron pump, Fig. 51. F is the valve-piston which drives the valve G, by means of a projection which sticks up through the neck of the

piston. The ends of the valve piston are hollow as shown in the cross-section of the right-hand end, and a small hole P, at each end admits steam at full pressure to the ends of the chest cylinder. If the main piston C is moving to the left (as it would be with the valve G in the position shown), when it reaches the end of the stroke, it will strike the reversing valve I′ and force it open, thus allowing the steam at the left of the valve piston to flow into the exhaust port through the passage E′. The steam at the right-hand end of the valve piston will expand, forcing the piston F to

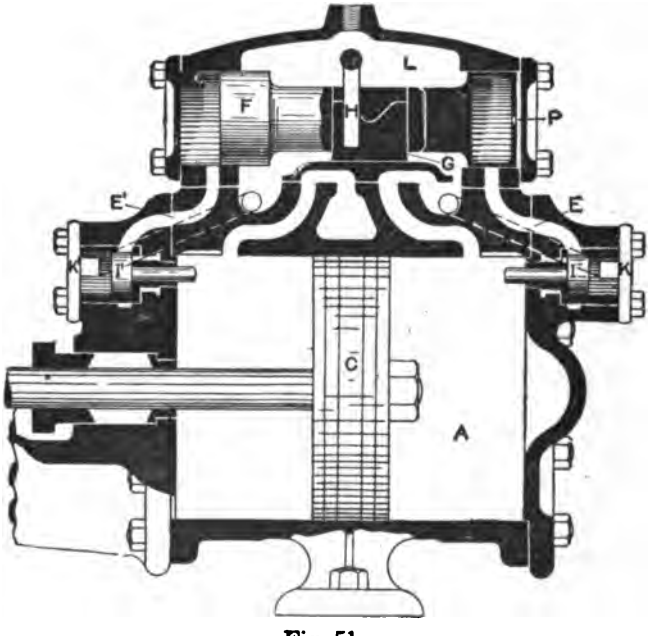

Fig. 51.

the left and carrying valve G with it, thus opening the right-hand end of the main cylinder A to the exhaust and the left-hand end to live steam, which will reverse the motion of the piston C. As the valve piston moves to the left, it will cover and close the port E′ hence will be brought to rest without shock by the cushioning of the enclosed steam. As soon as the piston C leaves it, the reversing valve I′ will be closed by the pressure in the passage K′ which is filled with live steam. At the other end of the stroke, the same series of operations follows by the action of the reversing valve I.

The valves require no adjustment and are easily brought to a new seat when they become worn, by removing the bonnets. If piston F becomes worn after long service, the cylinder may be re-bored and bushed or the holes P may be drilled a little larger, which will give greater pressure in the end of the chest cylinder and keep the pump working steadily for a long time even with

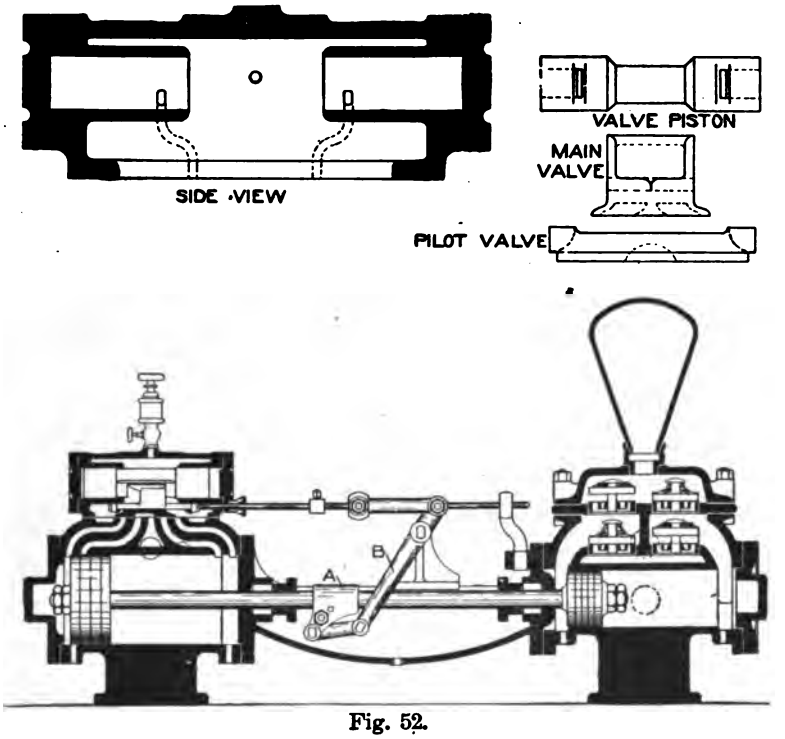

Fig. 52.

the piston F considerably worn. H is a starting lever, worked by a handle outside, for use in case the pump should stop with the valve G covering both admission ports.

A third style of valve is shown in Fig. 52, that of the Deane of Holyoke. In this the auxiliary valve is in the form of a yoke surrounding the main valve; the working-valve seat and ports for the auxiliary are at one side of the main-valve seat, and the ports supply steam to or exhaust it from the ends of the valve-piston

chamber through openings in the end of the piston. The auxiliary valve is moved by a valve rod actuated by tappets and a series of levers from the main piston rod. If the valve piston fails to move the main valve, the auxiliary valve, acting as a yoke, will finally drive the main valve over, by positive action.

A somewhat similar device is used on the Dean Bros'. pump, Fig. 53. There is, however, no period of rest of the auxiliary valve, the motion being similar to that given by an eccentric, and there is no provision for moving the main valve other than by the valve piston, because the positive driving of the auxiliary valve

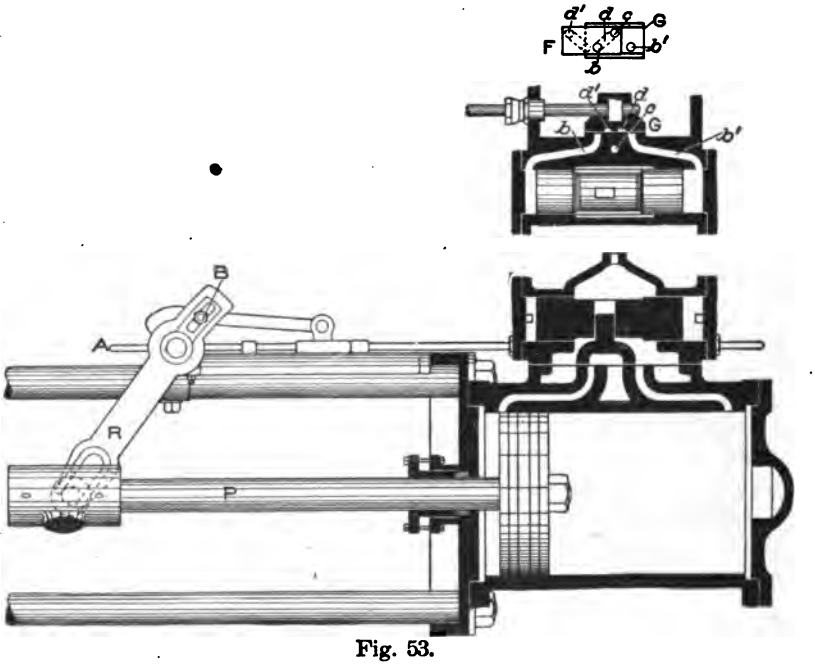

Fig. 53.

makes this unnecessary. The auxiliary valve F works on a seat G on the side of the valve-piston cylinder, being driven by the rod A and the rocker arm R from the main piston rod P. The length of its stroke can be changed by moving the pin B in the slot of the rocker arm, and the stroke of the main piston will thus be changed. The main valve is of the D form with a projection from the top which is engaged by the valve piston; the ports enter the cylinder, as in practically all direct-driven steam

pumps, at some distance from the end; thus steam will be caught to form a cushion for the piston at the end of the stroke.

The auxiliary valve, Fig. 53, is made very short and is so arranged that its ports are open only when the main piston is near the end of its stroke. Ports *b b'* are for admission of live steam and port *c* is for exhaust. All three ports are covered by the valve except when near the end of its travel; then the groove *d* in the face of the valve connects ports *b* and *c* and allows the steam to exhaust from the left-hand end of the valve piston while port *b'* is uncovered and live steam is admitted to the right-hand end; or, at the opposite end of the stroke, groove *d'* connects *b'* and *c*, and *b* is uncovered. This construction prevents waste of steam if the valve piston becomes worn in its cylinder, since live steam is conducted to that cylinder only during the moment of moving the piston. The stroke of the pump can be easily regulated and the action is noiseless.

A valve with a peculiar cut-off action is used on the Blake pump. In this, as it stands in Fig. 54, steam is entering the head end of the cylinder through the ports E and H, and is exhausting from the crank end through ports H', E', K and M. As the main piston A nears the crank end of the stroke, the valve C is moved to the left by a system of levers similar to those on the Deane pump, which strike tappets on the rod P. When this happens the lug S on the valve C covers the auxiliary steam port N, and S' uncovers the auxiliary steam port N' (see plan view), while the auxiliary exhaust port Z is disconnected from X' and connected to X, thus allowing steam to exhaust from the piston chamber at B' and flow into that at B. The main valve D will be carried to the right by the supplemental piston and will connect port E to K, leaving E' uncovered for the entrance of live steam, and the main piston will be forced towards the head end. It is not necessary to provide for positive mechanical driving of the main valve, as live steam can always enter one end or the other of both main and supplemental cylinders. The supplemental piston is fitted with rings so that it will take up its own wear, and prevent leakage into the exhaust, while the motion is such that the main valve has a slight lead on the main piston, thus admitting live steam to

cushion the latter at the end of its stroke and prevent slamming. Also, in case the main valve sticks, the valve C moves far enough to admit sufficient steam past the end of the main valve for the cushioning effect. The supplemental piston cushions on the steam caught as it passes by the end of the port X or X'.

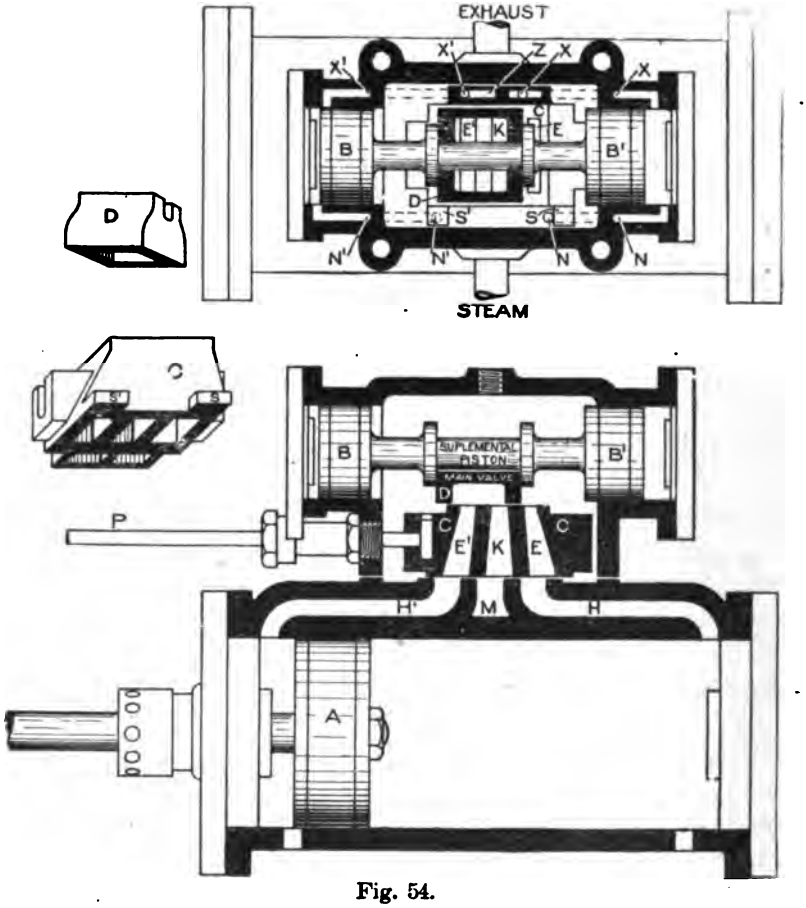

Fig. 54.

The Davidson pump, Fig. 55, has but one valve, operated by an auxiliary piston. The admission of steam to the ends of this piston is accomplished by the oscillating motion of the main valve itself. This motion is caused by a pin turned by a cam and oscillating lever driven from the pump rod. In the position shown, steam is being admitted to the crank end and exhausted

from the head end; consequently the piston is moving towards the right. As it nears the end of the stroke, the cam C will rotate the pin D and place valve A in position to connect the port E' with the exhaust chamber while opening E to live steam as seen

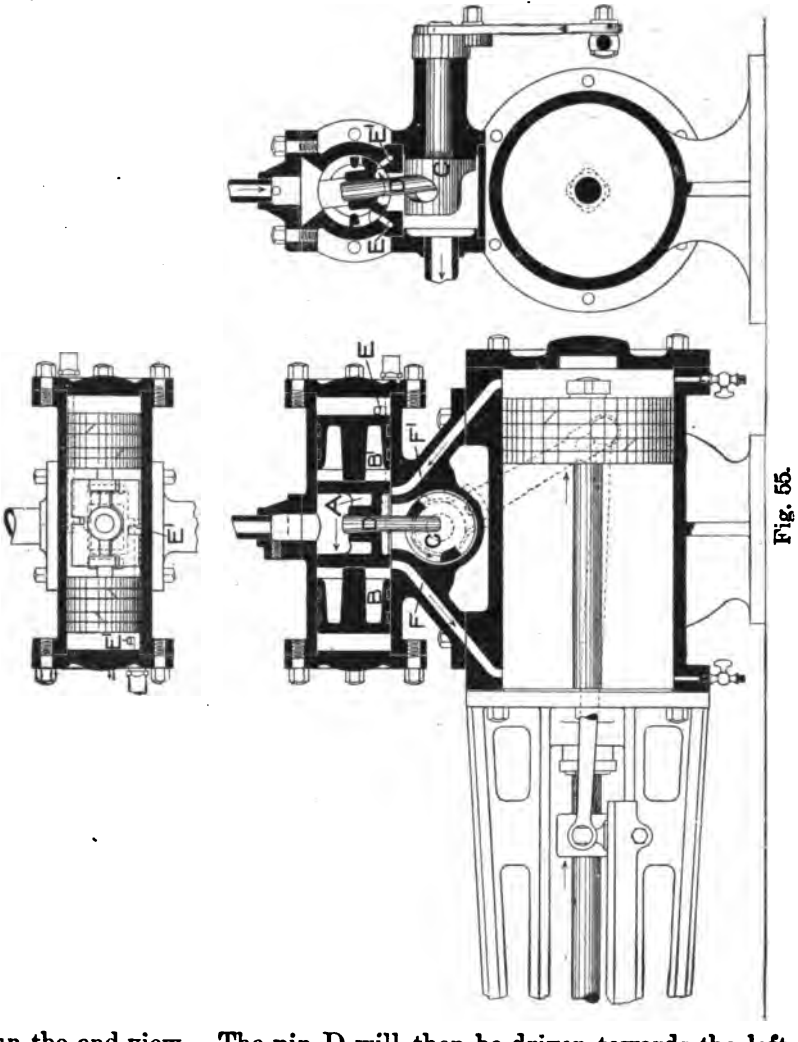

Fig. 55.

in the end view. The pin D will then be driven towards the left, carrying with it the valve A. Ports E' and F will be closed so that the main piston will be cushioned on the steam remaining in the cylinder.

The movement of valve A to this central position opens E′ to exhaust and E to live steam, and the piston B B′, carrying with it valve A, is forced to the left, opening port F to exhaust and F′ to live steam, and allowing the main piston to move towards the left When valve A is in a position to cover both ports F and F′, either E is open to steam and E′ to exhaust or vice versa, consequently the pump is always ready to start as soon as steam is turned on.

Various other forms of steam end have been used by different makers, but they work on practically the same principles as those described, being different only in details.

The duplex pump, as already explained, has two sets of cylinders side by side. The steam valve on one side is moved by

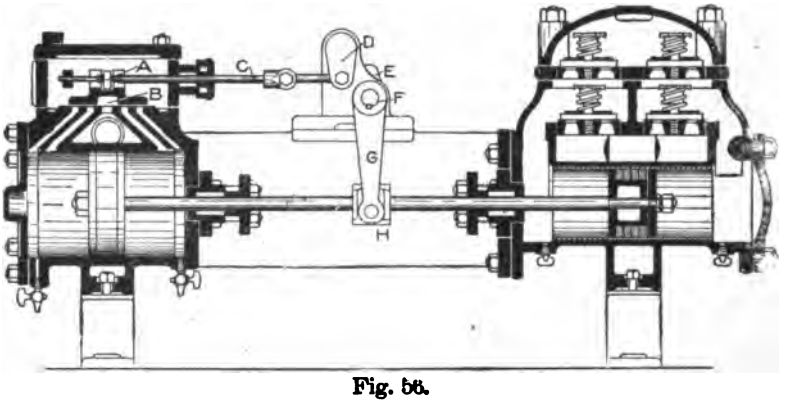

Fig. 56.

means of a valve rod and lever connected to the piston rod on the other side, so that one side starts when the other has reached about three-quarters stroke. The steam valves are usually of the D form, having separate ports for admission and exhaust at each end as seen in Fig. 56. In this figure, valve B is moved by rod C and bell-crank lever D, the hidden end of which is driven from the farther piston rod by a device like H, while the farther valve is driven by the rocker-arm lever G F E. The double ports are used so that the piston may run over the exhaust port and close it before reaching the end of the stroke, thus getting a steam cushion, and yet live steam may have a chance to enter behind the piston even if it stops quite at the end of its possible motion. In large sizes (particularly in fire pumps) a cushion valve is placed

in a passage connecting the steam and exhaust ports, so that the amount of by-pass opening between them may be adjusted, thereby regulating the cushioning to any desired amount.

In order to secure the rest period at the end of each stroke, which is needed for quiet running and small slip, it is usual to introduce a lost-motion device in the valve-driving mechanism. In Fig. 56 this is the nut A, which has a thickness less than the space between the lugs on the valve, thus preventing the valve from being moved until the nut has traveled some distance. In larger sizes it is more common to use a single lug on the valve; the rod passes through this and the lost motion is provided for by adjusting nuts held fast by jam nuts, so that the valve lug will have play between them as seen in Fig. 57. For very large pumps and pumping engines, lost-motion links are usually placed in the end of the valve rod, as seen in Fig. 58; this arrangement has the advantage that the amount of lost motion can be adjusted without removing the steam-chest cover, or even, if desired, while the pump is in motion. It is, of course, somewhat more expensive to construct than the jam nuts shown in Fig. 57, hence is not often used for small pumps.

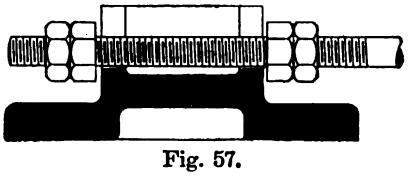

Fig. 57.

The amount of lost motion needed for any style of pump can be determined only by trial, but once fixed will be the same for all pumps of a given style and size. For pumps up to 10-inch stroke, $\frac{1}{8}$ to $\frac{3}{8}$ inch is usually allowed; for larger sizes the requirements call for $\frac{1}{2}$ to 1 inch.

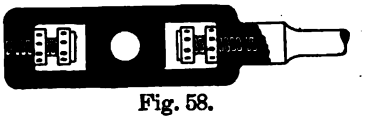

Fig. 58.

Compound Pumps. The simple steam pump must from the nature of its action take steam full stroke, hence has no possibility of using any of the expansive energy in the steam. In order to overcome this difficulty large pumps are often compounded, and usually with cylinders arranged tandem, as in Fig. 59, or with the smaller cylinder outside, as the designer may choose. Steam from the boiler enters the high-pressure steam-

chest in the usual way; it then passes to the high-pressure cylinder; from there it exhausts through the side-pipe, seen at the back of the steam-chests in Fig. 59, to the low-pressure steam-chest; thence passes into the low-pressure cylinder and from it to the exhaust pipe. Unless a boiler pressure of more than 80 pounds is used, it is not advantageous to compound a pump run non-condensing, as the saving in steam will not pay for the increased cost. If a condenser is used, compounding may be introduced with profit for pressures as low as 50 pounds; but the added complication of condenser and many cylinders is inadvis-

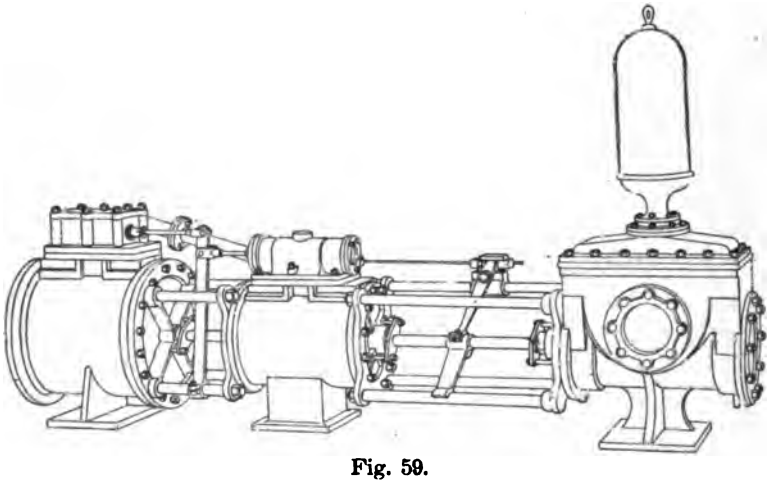

Fig. 59.

able for pumps with low-pressure cylinders smaller than about 18×24 inches.

Compound pumps are used either single as in Fig. 60 with an auxiliary valve and valve piston, or duplex with the valves driven by the method described for duplex pumps. As the pistons are on the same rod, the valves must move at the same instant, and are usually mechanically connected and driven by the same valve piston or lever, as the case may be. The duplex compound is more often used than the single, and it is considered better practice to use a duplex compound with small cylinders than to use a single compound with larger ones, as the difference in cost will be more than balanced by the steadiness of running

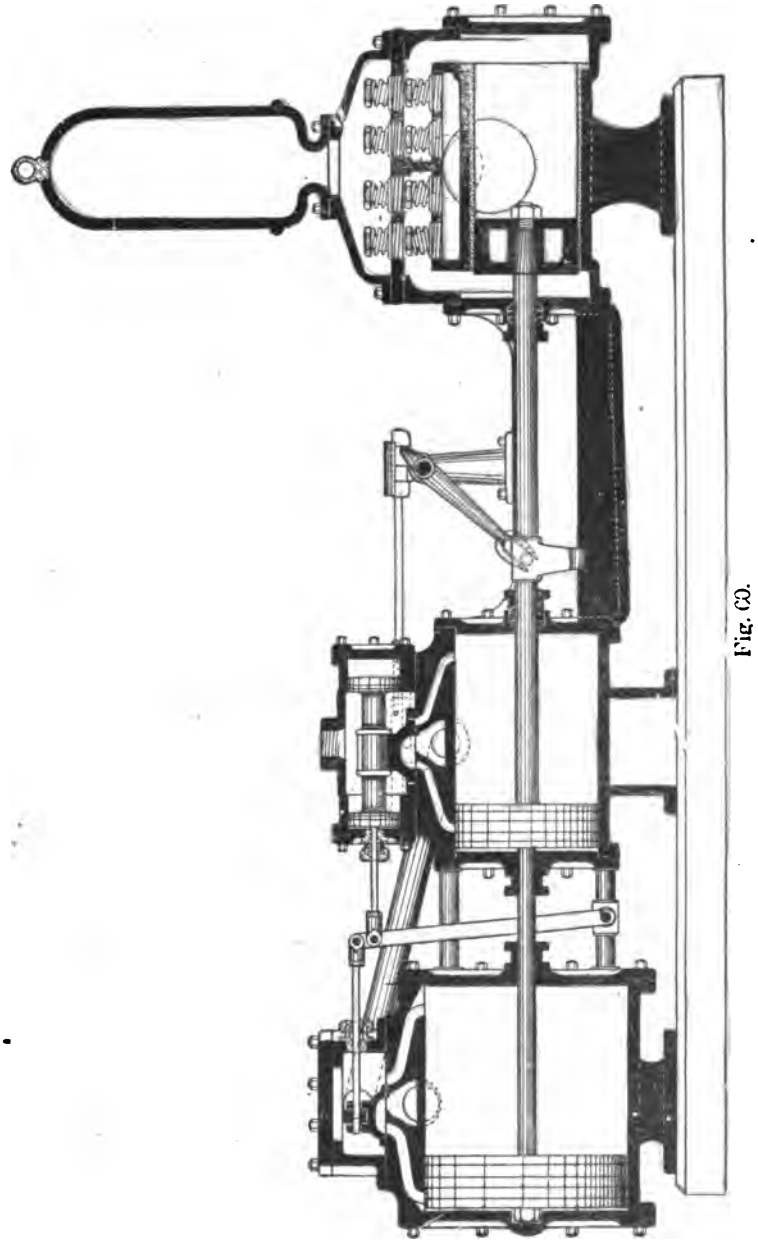

Fig. 69.

The high-pressure cylinder takes steam full stroke so that there is no expansion in it; at the opening of the high-pressure exhaust the pressure drops until it is equalized in the low-pressure steam-chest, side-pipe, high-pressure cylinder, and the clearance space of the low-pressure cylinder, the admission to which opens at the same instant as the exhaust from the high-pressure. On the return stroke of the pump, the steam expands in passing from the high-pressure cylinder to the low-pressure, the action being as indicated in the diagram, Fig. 61, and the nominal ratio of expansion between 2 and 3. In the diagram it is taken as 2.25, vertical distances being laid off to represent pressures and horizontal to represent volumes. The pressure in the passages beteen cylinders is a variable, but is taken as the value at the beginning of the low-pressure stroke. The back pressure in the low-pressure cylinder will be about 2 pounds above that at the outlet of the exhaust pipe, or 17 pounds for a non-condensing pump and 4 pounds for one run condensing.

The action illustrated in Fig. 61 is as follows: Assume that we are dealing with a pump whose high-pressure cylinder A is 16 inches in diameter by 18 inches stroke, and with clearance C 8 per cent of the piston displacement; and whose low-pressure cylinder B is 24 inches in diameter, the stroke being of course the same as for the high-pressure, and the clearance C 8 per cent of the low-pressure piston displacement. The intermediate space between cylinders, side-pipe and low-pressure steam-chest will be, in practice, about 0.35 the piston displacement of the high-pressure cylinder. Assume that the initial pressure is 85 pounds gauge or 100 pounds absolute and the back pressure in the low-pressure cylinder 4 pounds absolute. The line *a b* will represent the action during admission to the head end of the high-pressure cylinder, steam being taken at full pressure for the entire stroke. If we assume the volume of the piston displacement as 1, there will be 1.08 volumes of steam in the high-pressure cylinder and clearance. At the end of the stroke, the exhaust through the side-pipe into the low-pressure cylinder will be opened and the pressure will fall, the amount of the instantaneous drop depending on the pressure in the low-pressure steam-chest at the time the high-pressure exhaust opens. We shall see that this will be about 40 pounds, a

value which can be safely assumed as a working basis. When the high-pressure exhaust opens, the admission to the low-pressure cylinder also opens, and the pressure will drop according to Boyle's law, so that it is equalized in the three compartments.

The volume of the low-pressure cylinder will be $1.5^2 \times 1 = 2.25$, and of its clearance $.08 \times 2.25 = 0.18$ volumes. Then the pressure after drop and equalization is found thus:

$$\begin{array}{rrr} 1.08 \times 100 & = & 108.00 \\ .35 \times 40 & = & 14.00 \\ .18 \times 4 & = & .72 \\ \hline 1.61 & & 122.72 \end{array} \qquad \frac{122.72}{1.61} = 76.2$$

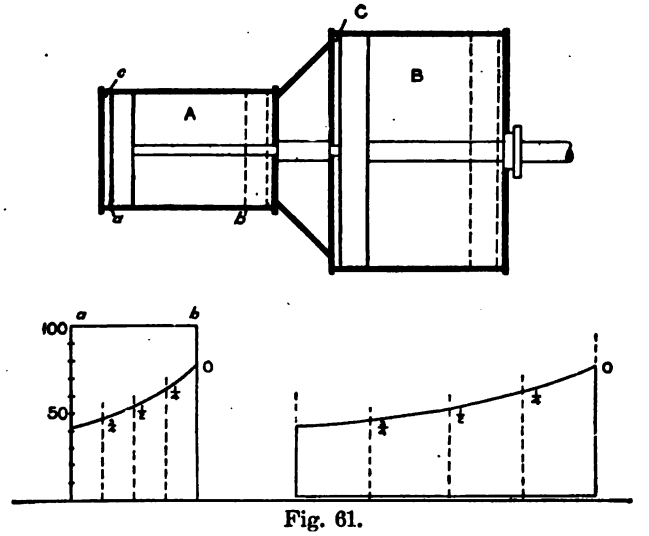

Fig. 61.

The pressure after equalization, 76.2 pounds, is found by multiplying each volume by the pressure existing within it and dividing the sum of the products by the sum of the volumes.

As the pistons return, the volume in the high-pressure cylinder decreases, that in the intermediate passages remains constant, and that in the low-pressure cylinder increases. The result will be an expansion according to the hyperbolic law so that the line showing the back pressure on the high-pressure piston, and that showing forward pressure on the low-pressure piston, will indicate the same pressure at each instant of the stroke, though the low-

pressure volume will be 2.25 times the high pressure. At $\frac{1}{4}$ stroke the total volume will be

$$.08 + .75 + .35 + .18 + (\tfrac{1}{4} \times 2.25) = 1.92 \text{ volumes,}$$

and the pressure $122.72 \div 1.92 = 64$ pounds.

Similarly at $\frac{1}{2}$ stroke the pressure will be 55 pounds,
at $\frac{3}{4}$ " " " " " 48.3 pounds,
at full " " " " " 42.9 pounds,

and so on for other points so that the whole expansion curve may be determined. In practice there will be a difference of about one pound between the back pressure in the small cylinder and the forward pressure in the large one, due to the friction of the steam in ports and side-pipe. On the next stroke, the head end of the high-pressure cylinder will take in a fresh charge of steam while the crank end of the low-pressure will exhaust its steam into the condenser. The crank end of the small cylinder and the head end of the large one act together in the same manner as described above, so that the pump is double-acting.

We are indebted to a paper on "Power of Compound Pumping Engines," by John W. Hill, published in *Engineering News*, for the proportions given in Table VIII:

TABLE VIII.

Ratio diameter l. p. cylinder.....			
to h. p. "	1.50	1.60	2.00
High-pressure cylinder volume taken as	1.00	1.00	1.00
High-pressure clearance volume.....	.063	.063	.06
Intermediate space volume.....	.339	.339	1.12
Low-pressure cylinder volume.....	2.25	2.56	4.00
" " clearance "	.112	.128	.204
No. expansions intermediate chamber..	1.319	1.439	2.250
No. expansions l. p. cylinder.....	1.825	2.020	2.258
No. expansions, total.....	2.400	2.907	5.081
Ratio Mean Effective pressure to (initial pressure minus back pressure).....	0.729	0.682	0.466

Setting the valves on a pump is for the most part a simple operation, and requires only that the valves shall be adjusted, usually by trial, until the pump makes the longest stroke possible without striking the heads, and reverses evenly at the ends of the stroke.

The fly-wheel pump is adjusted by moving the eccentric on

the shaft and the valve on its stem until steady running is secured, the same as for any slide-valve engine. The Knowles pump has the stroke lengthened by lowering the roll E, Fig. 50, and shortened by raising it; equalization of reversal is effected by lengthening or shortening the link L as may be required.

Adjustment of the Deane pump is made entirely by moving the tappets on the valve rod; the block A should be clamped to the piston rod in such position that when the piston is at mid-stroke the link B will be vertical. With the valve in mid-position the tappets should be placed so that they are equidistant from each end of the sliding collar, and then adjusted by trial until the working is satisfactory. If the piston strikes at either end, move the tappet at that end towards the collar; if reversal comes too soon at one end, move the tappet away from the sleeve at that end.

For the Dean Bros.' pump the same directions apply as to the Deane, but the stroke may be changed by moving the bolt B, Fig. 53, in its slot without changing the tappets.

The adjustment of the Blake valve is also the same as that for the Dean Bros.', but the tappets on the valve rod should be so set that the valve will have a little lead and open before the main piston reaches the end of its stroke. It has this lead by virtue of the action previously explained unless it is set to be very late in action.

In the Davidson pump, Fig. 55, the valve is adjusted for length of stroke by moving the bolt in the slotted end of the oscillating arm; shortening the leverage shortens the stroke and vice versa. To equalize the reversal, the sleeve which is clamped to the piston rod may be shifted towards the end at which it is desired to quicken the reversal, or a slight adjustment may be made at the point where the oscillating lever is made fast to the rock-shaft which drives the cam C.

For a single compound pump the valves are adjusted in the same way as for a simple pump of the same make. There is an adjustment in the connection between high- and low- pressure valves, and this must be set, with the steam-chest covers off, so that the main valves open at the same time.

All the valves of the duplex pump are set square so that, with

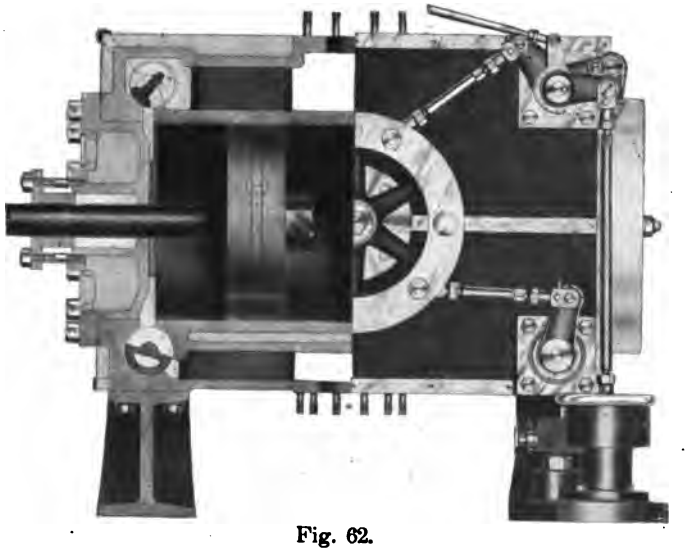

Fig. 62.

FULTON CORLISS CYLINDER.

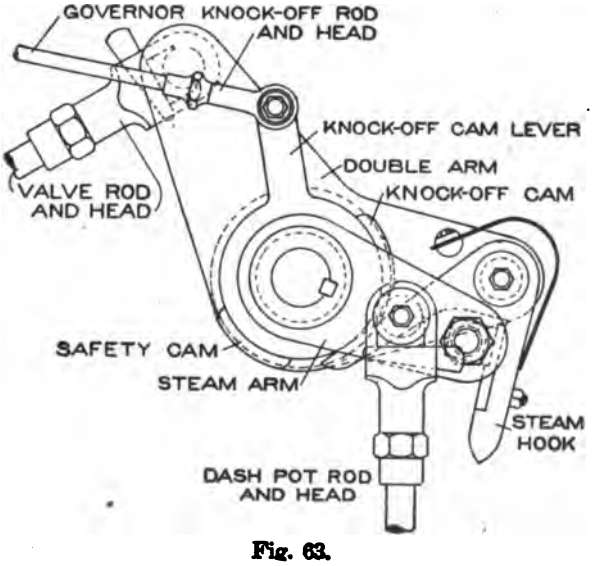

Fig. 63.

both pistons at the middle of the stroke, the valves will be in mid-position, the rocker-arm levers will be vertical and the lost motion will have equal play at each end. The amount of lost motion needed, where it is adjustable, can be determined only by trial, but generally should be as great as possible without having the pistons strike the heads. This will give a long stroke.

For the large pumping engines used in city water-works, the steam ends are designed the same as for any steam engine; and fly-wheels, main shafts and eccentrics are provided for steadying the motion, allowing of expansive working and operating the valves. For such engines the Corliss valves and valve motion have been most commonly used in the United States.

The arrangement of the motion and the action of the valves is shown in Fig. 62. The wrist plate is moved by the eccentric through a reach-rod, rock-shaft and eccentric rod, and from it links run to cranks which turn the rotating valves in their seats. The exhaust valves are positively driven, but the admission valves are so arranged as to be disconnected from the control of the link at some point in the stroke (depending on the position of the governor) by a trip motion, one form of which is shown in Fig. 63.

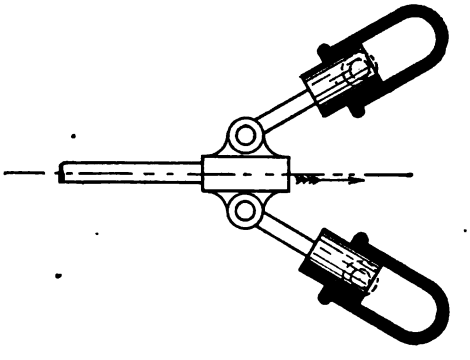

Fig. 64.

Another device used to allow of expansive working of steam in large pumps is the hydraulic compensator, Fig. 64. The steam must work against the pressure of the pistons up to half stroke, and is assisted by it beyond that point, so that cut-off may take place at half stroke, or later, and the energy of expansion be used beyond that point. With this device, there is no shaft or eccentric, and valves similar to those described for duplex pumps are used.

ERECTION AND PIPING.

The location of a pump should be chosen with two principal objects in view: To have the pump itself convenient for running, and accessible for repairs or adjustment; and to keep the piping as short and direct as possible. For a pump to which the liquid is raised by suction there is the limitation that it must not be placed more than 25 feet, and should not be more than 20 feet, above the source of supply; but aside from this, a matter of first importance is to have the machine where it will naturally be kept in good condition. Too often a pump is placed in a dark corner where it is never seen, seldom visited, and always neglected; it soon becomes dirty and leaky, decreasing its efficiency and shortening its life. The second point with respect to piping is often controlled by the layout of other apparatus quite as much as by the position of the pump itself; yet by careful study of the conditions it is often possible to find one place better than others for the pump. It is of more importance to avoid bends and elbows in water piping than in that for steam, because steam makes sharp turns with less friction loss than does water. If hot water is to be handled, the water should flow to the pump by gravity or under pressure. Otherwise the water will turn into vapor under the suction force and the pump will draw in either vapor alone or a mixture of vapor and water.

It is well to place a boiler feed pump in such a position that the gauge glass can be seen when standing at the pump, but this is not absolutely essential, and should be sacrificed if any gain in arrangement of piping or convenience of attendance can be secured thereby. If a condenser air-pump is so placed that the water from the condenser flows to it by gravity, it is possible to maintain a better vacuum in the condenser.

Wherever the pump may be located, a substantial foundation should be provided if the pump is of large size, especially if it is to be run at high speed. It is often sufficient to fasten a small pump to the floor or to heavy brackets secured to a wall, but a pump larger than a 4×6 should have a separate foundation. In designing the foundation, remember that the object is not so much to hold the pump up as to hold it down, to keep it from vibrating.

It is, therefore, better to have a foundation deep and narrow than broad and shallow, unless the pump is large and the soil very sandy.

The pump should be well bolted to the foundations in order to prevent vibration, as such movement not only is communicated to the pipe and thence to the building, but tends to loosen the joints in the piping and the pump itself. Absolute rigidity on the foundations should be secured at any cost.

The material may be stone, brick, or concrete, preferably the last. Stone is expensive and difficult to work; brick is liable, unless carefully laid with cement mortar, to be loose and lack compactness; while concrete is easily put down, is inexpensive and has all the solidity of stone. It should be made of good cement mortar, two parts sand to one Portland cement, mixed with broken stone not over 2 inches in longest diameter, in equal parts of stone and mortar. The concrete should be well compacted into a mould the shape of the foundation, the bolts being built in with plate-iron washers on the heads; the concrete should be deposited in layers about 6 inches deep and well rammed, a second layer being added before the upper surface has hardened. This process is repeated until the foundation is completed. It will generally be sufficient to finish the top with a surface of cement mortar carefully leveled and allowed to harden before setting the pump, but sometimes a cast-iron base-plate is used, and this gives a somewhat neater appearance.

As previously stated, the piping should be as short and direct as possible. In large work, the water pipe should have long-bend elbows and tees, and gate valves should be used to reduce the friction. Each pipe should be pitched throughout its length to one point so that it may be drained to avoid freezing; a drain-cock should be placed at the lowest point to remove the water. For water piping, it is well to use galvanized or brass pipe to avoid pitting or corrosion. Covering the pipes which carry cold water will prevent sweating and the consequent unpleasant dripping.

If the plant is one where a shut-down would be serious, a duplicate system should be installed, pump, piping and all; for any plant it is well to provide an injector as relay, in case the

boiler feed-pump will not work. In some cases duplicate piping is installed, but this seems hardly necessary, as piping is not likely to get out of order if well taken care of.

Wherever a long column of water is to be moved in either suction or delivery pipes, it is well to place a check valve near the lower end of the column to resist any tendency of the water to back up when the pump reverses or shuts down. In the suction system this valve would be placed on the inlet end of the suction pipe and is known as a foot valve, Fig. 65. For the delivery

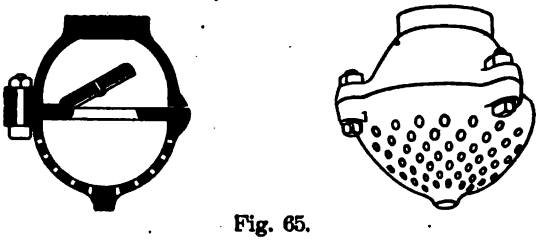

Fig. 65.

pipe, it is well to use a check valve near the outlet from the pump and another near the end of the pipe, especially if pumping against high pressure. The check valves may be of either flap or disc type, but if of the latter they should have ample area so that the double turn made by the current of water will not cause great loss of head. The flap valve is shown in Fig. 66 and the disc form in Fig. 67.

For all pumps which are to handle water from ponds, rivers or other sources where sticks, leaves, or any form of rubbish is likely to collect, a strainer as shown in Fig. 65 should be placed on the end of the inlet pipe. The combined area of the openings into the strainer should be 3 to 4 times the area of the pipe. As rubbish quickly collects on a horizontal strainer, the surface should be either slanting or vertical, and should be so designed that the screen may be easily cleansed. The straining surface should be fine or coarse in mesh according to the material to be screened, and made of woven wire or perforated metal as may be most con-

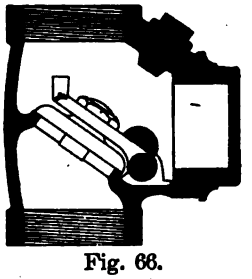

Fig. 66.

venient. Often the foot valve and strainer are combined into a single piece. If the lower end of the suction pipe is not accessible for cleaning, and if the debris is of such nature that it is likely to clog the openings, it will be better to use a design given by Barr (see Fig. 68) placed near the pump.

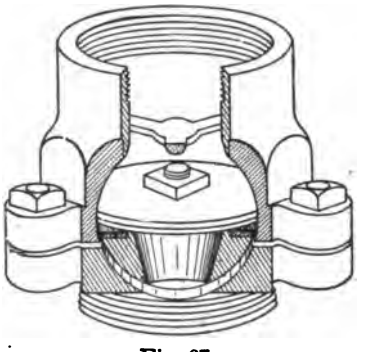

Fig. 67.

The sizes of pipe needed for the suction and delivery are determined by the pump maker and the pipes should never be made smaller than these; if the runs are long, the pipes may well be made larger.

The velocity allowable is, of course, the point which determines these diameters. For the suction, this is usually taken at 200 feet per minute or less; for the delivery pipe it may be 400 feet per minute or less.

The suction pipe should be of the same size throughout in order to avoid eddies and changes of velocity. Where the pipe is larger than the pump connection, the reduction should be made by a conical pipe with an easy taper, placed next the pump. The greatest care should be taken to see that all joints in the suction system are absolutely tight, as even a small leak greatly reduces the capacity and efficiency. The diameters of pipe suitable for direct-acting pumps are given in Table IX, the suction velocity being allowed at 150, and delivering velocity at 300 feet per minute.

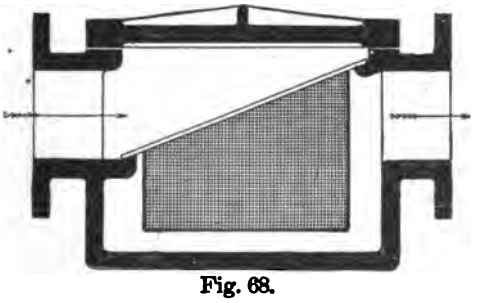

Fig. 68.

The loss in head depends on the length of the piping and the rate of flow of the water. Table X gives the loss in pounds pressure per 100 feet of pipe for various rates of discharge and sizes of pipe as stated by G. A. Ellis.

The loss of head from elbows and valves depends also on the

TABLE IX.

SIZES OF SUCTION AND DELIVERY PIPES.

Diameter Water Cylinder.	Diameter Suction Pipe.		Diameter Delivery Pipe.	
	Single Pump.	Duplex Pump.	Single Pump.	Duplex Pump.
4 inches	2½ inches	3½ inches	1¼ inches	2½ inches
5 "	3 "	4 "	2 "	3 "
6 "	3½ "	5 "	2½ "	3½ "
7 "	4½ "	6 "	3 "	4 "
8 "	5 "	7 "	3½ "	4½ "
9 "	6 "	8 "	4½ "	6 "
10 "	6 "	8 "	4½ "	6 "
12 "	7 "	10 "	5 "	7 "
14 "	9 "	12 "	6 "	8 "
16 "	10 "	14 "	7 "	9 "
18 "	12 "	16 "	9 "	12 "
20 "	12 "	17 "	9 "	12 "

Suction velocity, 150 feet per minute.
Delivery velocity, 300 feet per minute.
Piston speed, 100 feet per minute.

rate of flow, and is most conveniently referred to the length of pipe which would result in the same loss.

The resistance to flow of water in pipes due to bends, elbows, tees, etc., is stated by Foster in his "Electrical Engineer's Pocket Book," to be expressed by the equation:

$$P = F\frac{v^2}{64.4}$$

in which P is the loss in pressure in pounds per square inch, v the velocity of flow in feet per second, and F the coefficient of friction, which varies with the angle of the bend according to the following table:

Angle.	20°	45°	60°	90°	120°	135°
F	.020	.079	.158	.426	.806	.940

A globe valve will produce the same loss of head as two 90-degree bends, and a gate valve a loss equal to that from a 45-degree bend.

If water is known to contain lime or magnesia, it is certain that pipes will fill up more or less from the deposit of scale, and allowance should be made for this in the first place by using extra large pipe.

In choosing the size of steam pipe the same general principle applies; the exhaust pipe, like the suction, should be short, direct, and of ample cross-section. A velocity of 6,000 feet per minute is allowable in the steam pipe and 4,000 in the exhaust.

Care should be taken to have all piping which carries steam pitched away from the pump to avoid the collection of the water of condensation in the steam cylinder, and drips should be provided wherever there is any chance for water to collect.

The use of air chambers has already been discussed. They are usually a good investment if high speed or long pipe runs are to be used. The gain in durability and saving of repairs to the pump and piping system will more than pay the interest on the small cost of ample air chambers.

TABLE X.

FRICTION OF WATER IN PIPES.

Friction loss, in pounds pressure per square inch, for each 100 feet of length of different sizes of clean iron pipe discharging given quantities of water per minute. G. A ELLIS, C. E.

Gallons per Minute	SIZES OF PIPES—INSIDE DIAMETER														
	¾ in.	1 in.	1¼ in.	1½ in.	2 in.	2½ in.	3 in.	4 in.	6 in.	8 in.	10 in.	12 in.	14 in.	16 in.	18 in.
5	3.3	0.84	0.31	0.12	0.05	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
10	13.0	3.16	1.05	0.47	0.12	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
15	28.7	6.98	2.38	0.97	0.30	0.11	.....	.....	.....	.....	.....	.....	.....	.....	.....
20	50.4	12.3	4.07	1.66	0.42	0.15	.....	.....	.....	.....	.....	.....	.....	.....	.....
25	78.0	19.0	6.40	2.62	0.51	0.21	0.10	.....	.....	.....	.....	.....	.....	.....	.....
30	.....	27.5	9.15	3.75	0.91	0.33	0.11	.....	.....	.....	.....	.....	.....	.....	.....
35	.....	37.0	12.4	5.05	1.20	0.45	0.17	.....	.....	.....	.....	.....	.....	.....	.....
40	.....	48.0	16.1	6.52	1.60	0.52	0.22	.....	.....	.....	.....	.....	.....	.....	.....
45	.....	.....	20.2	8.15	2.00	0.65	0.28	.....	.....	.....	.....	.....	.....	.....	.....
50	.....	.....	24.9	10.0	2.44	0.81	0.35	0.09	.....	.....	.....	.....	.....	.....	.....
75	.....	.....	56.1	22.4	5.32	1.80	0.74	0.17	.....	.....	.....	.....	.....	.....	.....
100	.....	.....	.....	39.0	9.46	3.20	1.31	0.33	0.05	.....	.....	.....	.....	.....	.....
125	.....	.....	.....	.....	14.9	4.89	1.99	0.53	.....	.....	.....	.....	.....	.....	.....
150	.....	.....	.....	.....	21.2	7.0	2.85	0.69	0.10	.....	.....	.....	.....	.....	.....
175	.....	.....	.....	.....	28.1	9.46	3.85	1.00	.....	.....	.....	.....	.....	.....	.....
200	.....	.....	.....	.....	37.5	12.47	5.02	1.22	0.17	.....	.....	.....	.....	.....	.....
250	.....	.....	.....	.....	.....	19.66	7.76	1.89	0.26	0.07	0.03	0.01	.....	.....	.....
300	.....	.....	.....	.....	.....	28.06	11.2	2.66	0.37	0.09	0.04	.....	.....	.....	.....
350	.....	.....	.....	.....	.....	.....	15.2	3.65	0.50	0.12	0.05	0.02	.....	.....	.....
400	.....	.....	.....	.....	.....	.....	19.5	4.73	0.65	0.16	0.06	.....	.....	.....	.....
450	.....	.....	.....	.....	.....	.....	25.0	6.01	0.81	0.20	0.07	0.03	.....	.....	.....
500	.....	.....	.....	.....	.....	.....	30.8	7.43	0.96	0.25	0.09	0.04	0.017	0.009	0.005
750	.....	.....	.....	.....	.....	.....	.....	.....	2.21	0.53	0.18	0.08	.....	.....	.....
1000	.....	.....	.....	.....	.....	.....	.....	.....	3.88	0.94	0.32	0.13	0.062	0.036	0.020
1250	.....	.....	.....	.....	.....	.....	.....	.....	.....	1.46	0.49	0.20	.....	.....	.....
1500	.....	.....	.....	.....	.....	.....	.....	.....	.....	2.09	0.70	0.29	0.135	0.071	0.040
1750	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	0.95	0.38	.....	.....	.....
2000	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	1.23	0.49	0.234	0.123	0.071
2250	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	0.63	.....	.....	.....
2500	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	0.77	0.362	0.188	0.107
3000	.....	.....	.....	.....	.....	.....	...	.....	.....	.....	.....	1.11	0.515	0.267	0.150
3500	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	0.697	0.365	0.204
4000	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	0.910	0.472	0.263
4500	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	0.593	0.333
5000	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	0.730	0.408
Comparative Discharging Power of Pipes	$\sqrt{d^5}$	1	1.75	2.76	5.66	9.88	15.59	32.	88.2	181	316.2	498.8	733.4	1024.	1375.

Care should be taken, however, that no pockets are formed in the piping where air may collect, as the air cushion thus formed will serve no useful purpose, but will reduce both the capacity and the efficiency of the pump. The suction pipe should have a continuous rise from the source of supply up to the pump; and if an inverted U loop must be formed in the delivery piping it should have a pet cock inserted at the highest point so that whatever air collects may escape.

CARE.

After the valves of a pump are properly adjusted, the three things which ordinarily require care are: The lubrication, the packing, and the draining of cylinders. If these matters are carefully attended to, the pump will cause very little trouble.

For the smaller sizes the steam end is generally lubricated by means of a grease cup, which is filled with some form of tallow compound. As the heat of the steam melts this compound gradually, it flows into the steam-chest and is carried by the steam to the cylinder. For larger sizes, a regular cylinder oiler is used, or sometimes an oil pump driven by a lever from the main piston rod. In large plants, the pumps as well as the engines are fed from a central tank into which the oil is forced under pressure by a single large oil pump, and whence it descends by gravity to the various cylinders and passes through sight-feed oilers.

The stuffing box on the steam end usually gets sufficient lubrication from the cylinder, and the one on the water end gets water enough except when the packing is set up hard; then a little machine oil with graphite in suspension will help.

In the case of the steam-cylinder as for any other engine, the less oil used, so long as the piston works quietly, the better.

Flake graphite put into cylinder oil usually settles to the bottom of the cup, but if blown into the steam pipe so as to be carried along by the steam, it will work into the crevices in valves and piston rings and aid materially in reducing the oil required. Also if sifted over the packing when filling the stuffing boxes it will reduce the friction considerably.

For the bearings of valve motions, machine oil is, of course, used. These parts need the same care as any other machine bearings.

The packing in the water piston wears but slowly; nevertheless it should be regularly inspected to make sure that there are no leaks, as they would seriously impair the economy of the pump. Once in two months is not too often to examine the water pistons, and oftener should be the rule if there is any reason to suspect trouble. In repacking, or in tightening up either piston or stuffing boxes, there should be as little pressure as possible, above the limit to prevent leakage.

The piston ring packing of the steam end will wear for years if properly adjusted and lubricated. An inspection once a year is sufficient unless suspicious action in the cylinder seems to call for an investigation.

The stuffing boxes should be kept tight, but not screwed up so as to bind. It pays to be rather generous both in the amount of packing used in a stuffing box and in frequency of renewals; the former because a long bearing between rod and packing will keep tight with less pressure than a short one; the latter because old, hardened packing requires a heavy pressure to force it to a tight joint, and results in a large amount of energy wasted in overcoming friction. For stuffing boxes, any good square packing will answer, but the one on the water end is better filled with some form having a rubber compound for its main body, while that on the steam end works better with a flax packing, as the steam soon kills the rubber. Stuffing boxes should be refilled as often as twice a year, if the pump is in constant service.

The valves in the steam end have a sliding bearing, and will ordinarily wear to a true and tight joint. At the time of the yearly inspection, the head should be removed, steam turned on, and the valve worked back and forth by hand to make sure that no steam passes into the cylinder except when the admission valve is properly open. If the valves leak, they must be scraped to a bearing. Usually the valve face is scraped accurately by using a surface plate, and the seat is then scraped to fit the valve.

The valves in the water end may be reseated by grinding the seats and faces with tools made for the purpose, if both the valves and seats are of metal. If the valves have a bearing surface of composition, as is now the usual practice, the composition disc will wear rather than the seat, and the disc may be renewed easily and at

slight expense. Some makers use discs of soft alloy which are more durable than composition and also have its advantages.

In discussing the subject of capacity, the matter of speed was taken up. Manufacturers conventionally rate their pumps at 100 feet per minute, but this is not a good basis, and sixty double strokes seem to be more logical for computation. The object is to reduce slip and prevent pounding; hence, if special devices are used for opening and closing the water valves and to prevent slamming at the end of the stroke, there is no reason why a speed may not be used approaching that of power engines. The slip occurs almost entirely at the ends of the stroke and during the seating of the valves, so that it is always well to use a long stroke even though the diameter be somewhat small.

If mechanically operated valves together with high speed are used, ample air chambers should be placed on suction and delivery systems, as otherwise there is likely to be water-hammer due to the sudden stoppage of the column of fluid in the pipes when the valves close.

There is no difficulty with the steam end in using high speeds provided the steam is kept free from water. This necessitates, in the case of large pumps, a separator in the steam pipe just above the pump, to remove all water and ensure the passage of none but dry steam.

It is impossible to forewarn against all difficulties which may arise in running a steam pump, because it is always "something different" which happens, but trouble can often be traced to certain common faults. Defective valves in the water end and stoppage in the suction pipe are the probable causes for irregular working of a single pump. If the pump slams on one stroke and is steady on the other, it may be that the discharge valves are stuck open on one end either by friction or the lodgement of some substance on the seat. Often a jar with a hammer will remedy this defect, but it is advisable to take off the valve-chamber cover to find the reason for the sticking. If the slamming is on both strokes, it is generally due to stoppage in the suction pipe, or, in the case of a pump newly erected, it may be that the suction is too small. If the latter is the case, slowing down will stop it; but if there is stoppage, the pump will slam at all speeds. If the

suction is small, the addition of a suction air chamber will sometimes be beneficial. Slamming may also be due to a leak in the suction system in either valves or joints, or to a leak in the piston packing. Occasionally the springs on the inlet valves are too strong, though this is seldom the case. When starting up, air in the pump may cause it to slam, the remedy being, of course, to prime the pump and suction pipe by pouring in water, and to make sure, by opening the air cock on top of the water end, that all air is forced out of the valve chamber.

If a pump sticks at the end of the stroke, it is due to friction or improper valve setting. In the former case, relieving the pressure on the nuts which set up the glands to the stuffing boxes, until there is just enough pressure to prevent leakage, will overcome the difficulty. In the latter case the valve motion should be so adjusted as to act earlier in the stroke, but it is best to keep the stroke of the pump as long as possible in order to reduce the loss from clearance in the steam cylinder.

In starting up, particularly in cold weather, there will be considerable condensation in the cylinders, and water will form rapidly. This must be given a chance to work its way out through the drips which should be left open until the pump runs free and without sign of water in the steam; the warming up should be done at slow speed.

If a cylinder oiler is used it should be opened up a sufficient time before the pump is to be started, so that it may be ready to act immediately when the pump starts, as the lubrication is needed when the cylinder is cold, even more than at any other time. It is well to have a hand-forcing oil pump connected to the steam pipe of large pumps unless the feed of oil is by a positively-driven pump, so that in starting up, or in case of emergency, a supply of oil may be ensured.

Before starting, the suction and outlet valves should be inspected to make sure that both are open. If a start is made with these closed, it is likely to bring a pressure on the systems which will open the joints.

If the pump is a large one and is run condensing—that is, exhausting into a condenser—the condenser should be put in operation first by starting the flow of cooling water and the air pump.

When these are both working well and a good vacuum has been established, the main pump may be started.

In closing down, the lubricators may be closed a little before the time to stop. The drips on the steam cylinder should be opened after stopping in order to carry off the condensation from any steam which may remain. The drips on the water cylinder need not be opened unless there is danger of freezing, in which case the whole water system should be drained. For this purpose the piping should all pitch toward the pump so that the water may all be drawn off at that point.

In the duplex type of pump, unless the packing be adjusted with even pressure on both sides, the side on which the tighter adjustment is made is liable to "short stroke;" in fact this is usually the trouble with a pump which goes "lame" on one side. If the short stroking is on one end only, it is probably due to poor setting of the valve-motion tappets. Short stroking may also be due to tight packing on the water piston, in which case it can be remedied by taking out the packing and cutting off a little. This is likely to occur only with pistons packed with square packing, as those having the cup leather packing or piston rings adjust their own pressure between piston and cylinder.

In setting up a new pump, it is important to blow out all pipes before making the connections, in order to make sure that no chips or dirt get into the pump. Unions should be provided on each pipe near the pump, so that, in case of suspected stoppage of the pipe it can be readily inspected.

TESTING.

The power used by a pump is usually so small an item in the running expense of a plant, that a test is considered unnecessary; and also the steam exhausted is often used for heating feed water or for some industrial process. If this is possible it is usually an economical way of running; but in plants where many pumps are needed, and where heating feed water is the only use for the exhaust steam, more steam will be available than can be used to good advantage; this is especially true if the main engines run non-condensing. An economical pump is then of great importance.

Duty. In order to determine the good or bad performance of a pump, it is necessary to test it for efficiency and slip, the

test being known as a "duty trial." The duty of a pump, as the term descended to us from the days of Watt's early pumping engines, was the number of foot pounds of work produced by 100 pounds of coal. This is a convenient basis for comparison of engines, but is not accurate, as coal varies so much in heat value; also this method of reckoning involves the efficiency of the boiler as well as that of the pump. As an attempt to eliminate the latter source of error in making comparisons, a conventional assumption has been made of 10 pounds of steam evaporated per pound of coal; but this is really only the substitution of one error for another, for a pound of steam when measured in heat units is by no means a constant, and a pound of coal seldom does evaporate 10 pounds of steam in actual boiler performance. The more logical method of comparison is by the duty per 1,000,000 heat units furnished to the pump, a basis proposed by a committee of the American Society of Mechanical Engineers appointed to formulate a code for conducting such tests.

The duty of the pump is found by measuring the quantity of water delivered and the height through which it is lifted, or its pressure equivalent. The coal or steam used must also be measured. Then

$$\text{Duty} = \frac{Q\,H \times 100}{C}$$

in which Q = pounds of water delivered,

H = head against which the pump works, both suction and forcing, and

C = pounds of coal burned.

Or on the new basis,

$$\text{Duty} = \frac{Q\,H \times 1{,}000{,}000}{B\,T\,U}$$

where Q and H are as before, and B T U is the heat units in the coal or steam, whichever is measured.

An inaccurate method of computing duty is sometimes used, which is based on the area of the water piston or plunger and its travel; but this takes no account of the slip, which may in small pumps be as much as 20 per cent; hence it is to be condemned.

The slip is found by comparing the water actually delivered with the total piston or plunger displacement, for the time of the

test. The difference, measured as a per cent of the piston displacement, is the slip.

The water delivered is measured either by weighing, by the use of calibrated tanks, by a weir, or by some form of meter. One of the first two methods is best for small pumps, and the meter is practically the only means available for large ones where the delivery is into a closed pipe system. For the weight or tank method, the arrangement of Fig. 69 is used, one tank being filled while the other is being emptied, through a quick opening gate valve. If the water is weighed, a certain amount as nearly as may be is run into the tank and the exact weight is caught after the valve is closed and the water is flowing into the other tank. For calibrated tanks, the filling is done nearly to a set mark, at which

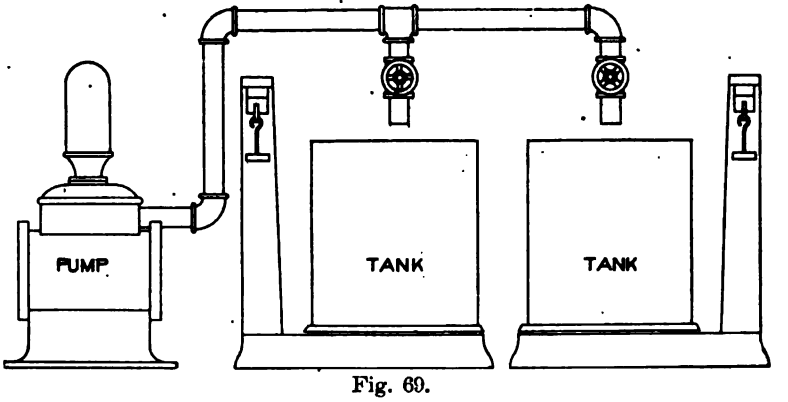

Fig. 69.

the capacity of the tank is known, and the exact level is found by dipping from the second tank. The weir can be used only where the delivery is into an open vessel, as the current of water must be made to flow over an open notch as in Fig. 70. The amount of water which passes this notch evidently depends on its length, and on the head of water above the sill. There is a certain contraction as the water enters the notch; but if the edges are beveled to a sharp edge up stream as shown, this will be slight. The depth of the notch should be not over $\frac{1}{2}$ the length, and is better made considerably less, say from $\frac{1}{10}$ to $\frac{1}{4}$. The over-fall below the notch should be at least twice the depth of the notch. The head of water over the sill should be measured at a point some distance back of the notch in order to get a quiet, even surface.

Any method of measuring the head will answer which gives it accurately; but a hook gauge, such as shown in Fig. 71, will give the best results. The reading of the gauge should be taken when set so that the point just breaks the surface as it is brought up from below, when no water is flowing over the notch. This gives the height of the sill from which to calculate. When the water is flowing from the pump, a reading from the gauge is again taken as the point just breaks the surface. The difference between the two readings is the head above the sill.

The flow, if there were no end contraction, would be $Q = b\,v\,h$ where Q is the cubic feet per second, b the length, h the head of

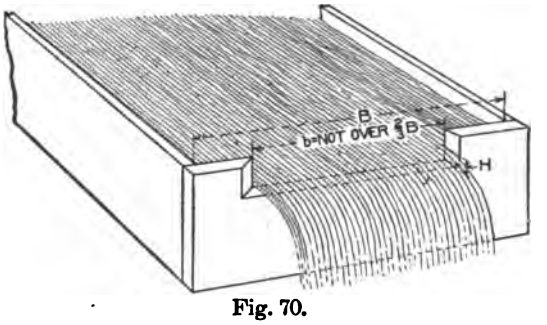

Fig. 70.

water above the sill, and v the velocity in feet per second. But $v = \sqrt{2\,g\,\frac{h}{2}}$ the same as for falling bodies, $\frac{h}{2}$ being used because it is the head of center of flow. It has been found by experiment that there is a contraction at the ends and bottom of an opening, due to the in-rush of the water from all sides, and that the flow will be about .62 the theoretical amount; hence

$$Q = .62\,bh\sqrt{2g\,\frac{h}{2}}$$

$$= .62\,bh\sqrt{gh}$$

$$= .62\,bh^{\frac{3}{2}}\,g^{\frac{1}{2}}$$

but g is the acceleration due to the force of gravity and is equal to 32.16; hence $.62 \times g^{\frac{1}{2}} = 3.52$ and $Q = 3.52\,bh^{\frac{3}{2}}$.

It is found, however, that for accuracy the formula must be modified to take account of the depth of the water, and also that the coefficient is too large. Smith gives, in his Hydraulics, the equation

$$Q = 3.29\,(b + \frac{h}{7})\,h^{\frac{3}{2}}$$

which is accurate for weirs for depth h, from 6 inches to 2 feet and with length b, not less than $3h$. For large quantities of water which must be delivered under pressure, the Venturi meter is the most accurate means of measurement. This is a patented device manufactured by the Builders Iron Foundry of Providence, R. I., and registers the flow by means of a recording mechanism driven by clockwork. The next best device is some form of rotary water meter, carefully calibrated. These will work well with cold water, but are not to be relied upon with hot water as the varying temperatures affect the readings appreciably.

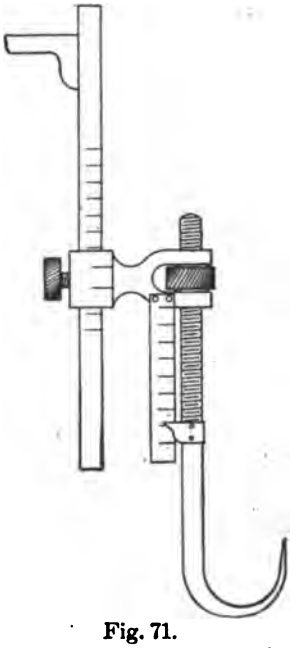

Fig. 71.

For methods of testing large pumping engines, the student is referred to the report of the committee of the American Society of Mechanical Engineers, Vol. XI of the Transactions, which can be obtained from the Society in pamphlet form at nominal cost.

The following discussion will be confined to small pumps :

For a power-driven pump it is necessary to measure the power supplied. If belt-driven, this can best be done by means of a transmission dynamometer, which measures the pull on the belt. Then this pull in pounds times the speed of the belt in feet per minute equals the foot pounds per minute; and that quantity divided by 33,000 gives the horse power. The work performed per minute will be the weight of water pumped per minute times the distance through which the water is raised as indicated by gauges on the suction and delivery pipes near the pump. These heads can be derived from the ordinary pressure gauge readings

by the table on page 9. The work per minute divided by 33,000 gives the delivered horse power. For a motor-driven pump the power applied can readily be obtained by electrical measurements. Measure the voltage and amperes of current with the pump carrying its load and then Voltage $\times \frac{\text{Current}}{746} =$ Horse power. From an efficiency curve of the motor as supplied by the makers, or obtained by a dynamometer test of the motor, get the efficiency at the horse power thus found. Multiply the horse power by the efficiency and the product will be the power supplied to the pump.

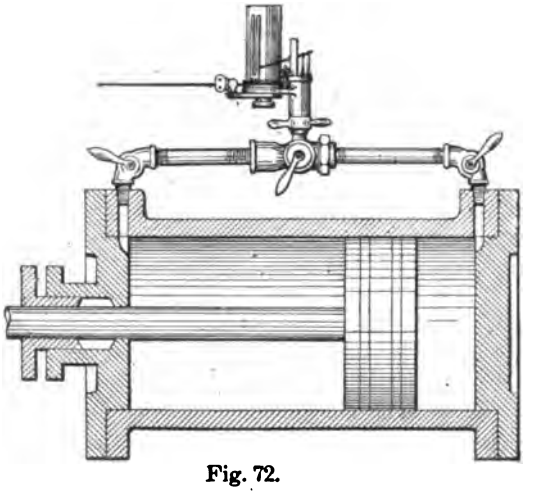

Fig. 72.

The power delivered by it in the form of water pumped is found by the same method as for the belted type.

For a steam-driven pump it is necessary to attach an indicator to the steam end in order to measure the power supplied by the steam. The indicator is connected as shown in Fig. 72, to register the pressure in the cylinder at each point of the stroke. The average of these pressures is then found, measuring between the lines indicating the steam and exhaust pressure as shown in Fig. 73. The sum of lines 1, 2, 3, 4, etc., divided by the number of lines gives the average length, and this multiplied by the scale of the spring, or pounds pressure represented by an inch of height, will give the average effective pressure. The scale of the indicator spring is always found stamped on the cap at one end

of it. This average pressure multiplied by the area of the piston in square inches, by the length of the stroke in feet and by the number of double strokes made per minute, gives the work done per minute in one end of the cylinder. In the same way pressure times area, times length, times number of strokes, gives the work for the other end, and the sum of these amounts divided by 33,000 gives the horse power developed. It should be noted that there is a difference between the areas of the two ends of the piston owing to the insertion of the piston rod in one end.

Sometimes the work done in the water cylinder is measured by using the indicator in the same way as described for the steam end. The work done by the water piston can, of course, be found by this method; but there is nothing to show whether the work is used in pumping water or in slip and leakage, except that a slow

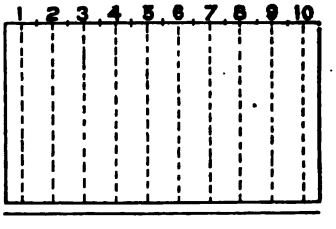

Fig. 73.

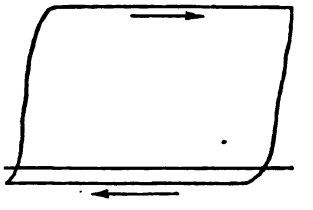

Fig. 74.

seating of the valves may show in a reduced average pressure from the diagram, as in Fig. 74.

To test the leakage past the piston, one cylinder head may be removed and the pump run single acting, the water which passes the piston into the open end of the cylinder being caught and weighed. This does not, however, measure the leakage due to faulty valves or seats, and the only way this can be found is by measuring the water actually pumped.

The slip is then found as follows: Multiply the area of the water piston by its length of stroke and by the number of single strokes if single acting, or double strokes if double acting—all dimensions being taken in feet—to get the number of cubic feet of water which would be pumped per minute as obtained by measurement; subtract the water actually pumped and divide the remainder by the computed volume which should be delivered.

The quotient expressed as a percentage is the slip. Slip is due to the leakage past the piston plunger and through the valves, and the amount varies with the condition of packing and seats and with the promptness of valve closure. It can be kept at a minimum by careful attention to packings, and by running slowly and steadily to allow the valves time to seat.

The efficiency of the pump may be expressed according to various standards. If efficiency as a machine, or mechanical efficiency, is desired, it is found by dividing the horse power utilized in pumping water, by that furnished to the pump by belt, motor or steam cylinder. If efficiency of the water end is wanted, it is found by subtracting the percentage of slip from 100. The total efficiency is the mechanical efficiency multiplied by the pump efficiency.

To find the "duty" the work done in a given time is found by one of the processes already indicated; the heat furnished during that time is computed from the coal burned and the efficiency of the boiler, or better, by condensing the steam used in a surface condenser, weighing it and calculating the heat needed to evaporate that amount of steam, starting with water at the temperature of the exhaust steam. Reduced to the form of equations this becomes:

$$\text{Duty} = \frac{\text{Foot pounds of work done.}}{\frac{\text{Heat units used}}{1{,}000{,}000}}$$

Foot pounds work = weight of water multiplied by equivalent head overcome. Heat units used = Pounds of steam used × (total heat at initial pressure minus the heat of the liquid at exhaust pressure).

The values inside the parenthesis must be obtained from tables of the properties of saturated steam, which are given in "Boiler Accessories," works on the steam engine, or engineering handbooks.

For a power-driven pump the duty is found by the formula:

$$\text{Duty} = \frac{\text{Foot pounds of work done.}}{\frac{\text{Foot pounds furnished the pump.}}{778 \times 1{,}000{,}000}}$$

since 778 foot pounds are equivalent to one heat unit.

For a motor-driven pump, the foot pounds are equal to average volts × average amperes × time in hours × 2,654.2.

PUMPING BY COMPRESSED AIR.

A system of pumping water by compressed air was known early in the nineteenth century, but no practical working apparatus was devised until Dr. J. G. Pohlé took up the matter in the early seventies.

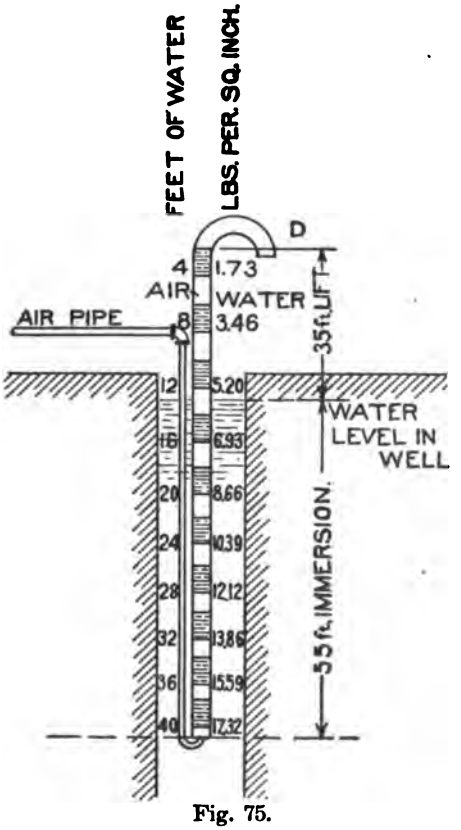

Fig. 75.

The principle of operation is shown in Fig. 75. A discharge pipe, D, is let down into the well so that its lower end is deep below the surface. Compressed air is forced down through a smaller air pipe, and is liberated inside the lower end of the discharge pipe at a high pressure. The air thus set free forms a big bubble the full size of the interior of the discharge pipe, displacing an equal volume of water, and thus making the weight of the column of water and air inside the pipe less than an equal volume of water outside the pipe. The column inside the pipe will therefore rise; and since the formation of air bubbles is continuous so long as the compressed air is supplied, a stream of slugs of water and bubbles of air will rise through the discharge pipe and flow out at the upper end.

The air pressure used must at first be sufficient to overcome the head of water in the discharge pipe, as well as the friction against the sides of the pipe. Also the stream of water must leave the discharge pipe with some considerable velocity.

The pressure needed bears a ratio to the pressure due to the total head of lift and immersion varying from 0.77 for comparatively shallow wells (100 feet or so deep) yielding large volumes of water, to 0.62 for wells 500 feet in depth and having but a small flow. Obviously the lowest air pressure which will do the work is the most economical, since any excess of pressure is used up in producing unnecessary velocity of discharge at the outlet.

The efficiency increases as the ratio of submergence of the lower end of the discharge pipe below the water-level to lift above the water-level increases. There is much disagreement as to the efficiencies that can be obtained; but tests seem to indicate that an efficiency ranging from 18 per cent (for a ratio of 1.5 submergence to lift) up to 37 per cent (for a ratio of 2.6), the efficiency being based on the power required to compress the air, is as much as can be expected.

A great deal depends on the proper proportioning in size of air pipe to discharge pipe, and of submergence to lift. A cross sectional area of water pipe $6\frac{1}{4}$ times that of the air pipe has been found satisfactory; and a submergence of twice the lift is common in practice, though some tests seem to indicate that a ratio of submergence to lift of 3 to 1 would give better results. The ratio should not in any case be less than 1.5 to 1.

In this connection it should be remembered that the water-level when pumping will always be lower than when at rest, and the above ratio should be fixed for the level under working conditions.

If the air pipe be too large, more air will be furnished than is needed and the discharge velocity will be too great; if the air pipe be too small the air bubbles will not expand sufficiently to fill the discharge pipe but will rise through the water without pushing it along upward.

A velocity not exceeding 20 feet per second is recommended in the air pipe.

The machinery necessary for an air-lift system comprises the following: an air compressor of size sufficient to give the needed air supply, a reservoir large enough to break up the pulsations from the compressor and steady the flow, gauges and valves for regulating the pressure, and the necessary well piping.

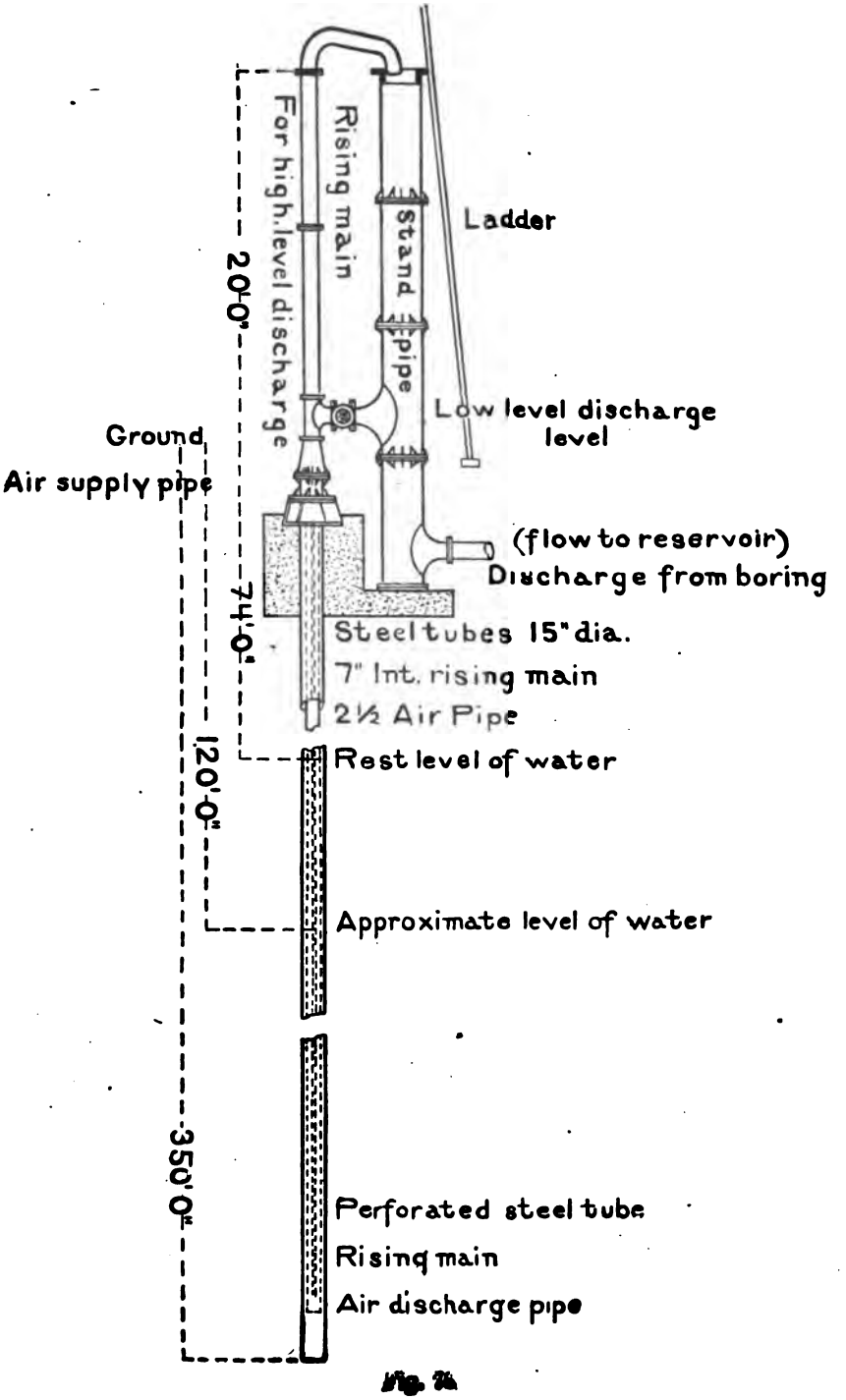
For high level discharge
Rising main
Stand pipe
Ladder
20'0"
Low level discharge level
Ground
Air supply pipe
(flow to reservoir)
Discharge from boring
74'0"
Steel tubes 15" dia.
7" Int. rising main
2½ Air Pipe
120'0"
Rest level of water
Approximate level of water
350'0"
Perforated steel tube
Rising main
Air discharge pipe

Fig. 24

For pressures up to 60 pounds, a single-stage duplex compressor will answer; for pressures between 60 and 300 pounds a two-stage compressor, with an inter-cooler between cylinders, will be found economical; while for pressures above 300 pounds a three-stage compressor should be used.

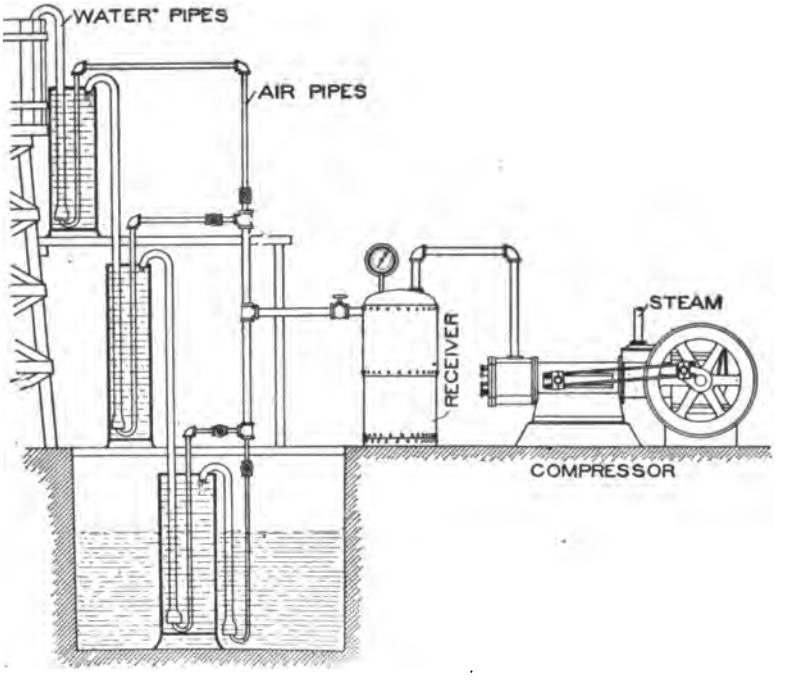

Fig. 77.

The supply of air needed will vary with the lift, and of course with the quantity of water handled. Wm. H. Maxwell gives the following equation:

$$V = \frac{G \times L}{20}$$

where V = cubic feet of free air per minute.

G = cubic feet of water per minute.

L = lift of water in feet.

The arrangement of air pipes and discharge pipes is largely one of convenience. The least friction will be produced by admitting the air to the bottom of the discharge pipe as shown in Fig.

75; but in deep wells this is often inconvenient, and the air pipe is carried down inside the water pipe as in Fig 76.

For high lifts it is sometimes convenient to use multiple-stage arrangements as shown in Fig. 77, the object being to secure sufficient depth of immersion without the expense of drilling a very deep hole. Only one air compressor is needed, as the pressure can be controlled by throttling the valves for the lower stages without serious loss of economy.

The advantages of the air-lift system are its simplicity, its concentration of all machinery into one place which can be conveniently located, and its ability to handle water containing grit, stones or ashes without injury to the machinery. On the other hand the efficiency is low, and a great depth of well in proportion to the lift is needed. The air-lift system is not adapted to all classes of service, but for handling a number of scattered wells, or a single deep well in an awkward location, it is simple and effective.

EXAMINATION PAPER

STEAM PUMPS.

Read carefully. Place your name and full address at the head of the paper. Any cheap, light paper like the sample previously sent you may be used. Do not crowd your work, but arrange it neatly and legibly. *Do not copy the answers from the Instruction Paper: use your own words, so that we may be sure that you understand the subject.* After completing the work add and sign the following statement:

I hereby certify that the above work is entirely my own.

(Signed)

1. Find the pressure per square inch corresponding to a head of 187.42 feet.

2. Under what conditions should a centrifugal pump be used?

3. Describe the rotary pump.

4. If a large quantity of water is to be lifted a slight amount (for instance 20 feet) is it better to use a direct-acting pump or a centrifugal pump?

5. How many gallons per minute can be raised by a pump having a water cylinder 8 inches in diameter, 12 inches stroke, if it makes 45 double strokes per minute? Assume 6% slip.

6. What materials are used for valve discs for cold water? For hot water?

7. Which is the better form of spring for a valve, cylindrical or conical? Why?

8. Under what conditions is it better to use a plunger pump rather than the piston type?

9. A single cylinder pump is 6"x10". What should be the volume, diameter and height of the air chamber on the discharge pipe?

10. A 12" x 10" duplex makes 52 double strokes per minute. What is the piston speed in feet per minute and what is the discharge in gallons?

11. A double-acting pump is to discharge 232 gallons per minute. On account of the pressure against which the pump must work, the water cylinder must be 7 inches in diameter. The allowable number of double strokes per minute is 55. What is the length of stroke? What is the piston speed?

12. Suppose a pump has a suction of 14 feet and the loss due to friction in the piping is 3 pounds per square inch. The pump is a 3″ × 10″ and the discharge is against a head of 86 feet. If the pump makes 40 double strokes per minute, what horse power is required, allowing 20% for friction?

13. The head against which a pump works is 122 feet. If the plunger is 4 inches in diameter and the steam end 7 inches, what steam pressure must be used? Consider the plunger friction to be 75 pounds.

14. A boiler-feed pump works against a pressure of 145 pounds. It draws water from a tank located so that the lift is 16 feet. If the steam pressure at the pump is 130 pounds and steam cylinder is 8 inches in diameter, what is the diameter of the plunger? (a) Make no allowance for friction. (b) Assume friction to be 15%.

15. Describe the setting of the valves of a duplex pump.

16. Describe the valve motion of a duplex pump.

17. Draw a diagram and locate the high-pressure cylinder, the low-pressure cylinder, the pistons, the piston rod, water cylinder, plunger, and air chamber of a tandem compound pump.

18. What means are used for connecting the shaft of a gas or steam engine to a triplex pump?

19. How is steam cushion obtained in a pump?

20. Why are laps added to steam valves?

21. What should be the maximum lift for a disc valve?

22. Explain with sketch the action of the combined forcing and lifting pump (double acting).

23. Describe the usual lost-motion device for a duplex pump. Why is it used?

24. Where should the check valve of a suction end be placed?

25. What kind of packing is commonly used in the water-end stuffing box?

26. If a pump slams on one stroke but not on the other, what is the probable cause?

27. Give method of procedure in starting a compound condensing pump.

28. What is the most common cause of short stroke on one side of a duplex pump?

29. What is meant by "duty"?

30. How may the slip be found?

31. The statement on page 42, lines 8, 9, and 10, gives the ordinary method of determining the actual from the theoretical H. P. Is it correct mathematically? What is the actual efficiency assumed in the case?

32. What size rod would you use for a 9" piston?

33. What would be the diameter of the valves for a 10" x 12" pump, if four valves were used?

34. What form of steam valve is used for direct-acting pumps? How are the valves operated?

35. What are the important points to watch in piping up a pump?

36. Where could graphite be used to advantage in pump running?

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